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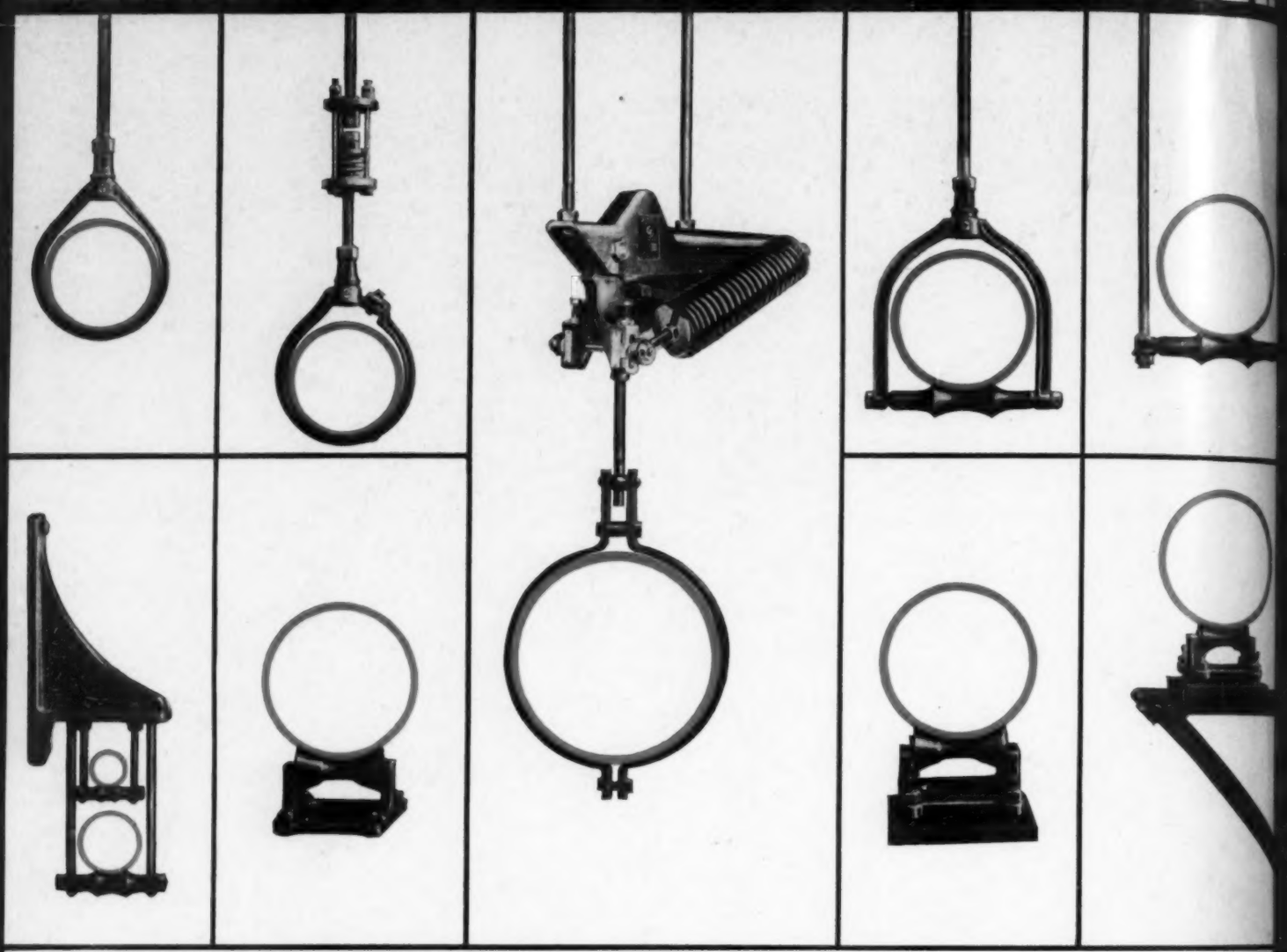

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The Editor's Page

■ **On a par** with telephones, good tools and desks, air conditioning is recognized as an essential part of working equipment in most business houses today, according to an editorial in "The Business Conditioner," official publication of the Air Conditioning Bureau of Omaha. In estimating the lost time which could be eliminated by proper air conditioning, "The Business Conditioner" says: "The average office worker spends at least 15 minutes per day adjusting windows and radiators in winter and trying to cool off in summer—including the time wasted in concentrating on the work at hand after the interruption. This amounts to more than sixty hours per year per employee. . . . Time lost through absence from work averages seven hours per year per worker. Records show that, of this lost time, 75 per cent is due to colds, coughs, grippe and other respiratory diseases. In two large offices studied, year 'round air conditioning resulted in a reduction of 50 per cent of the absences due to these causes. The saving thus effected through reduced absence paid for one-half the cost of owning and operating an air conditioning system. Increased efficiency was found to easily offset the remaining half of the cost."

■ **Some 450** persons registered for the conference on air conditioning and automatic heating held at the University of Wisconsin last month. . . . A feature of the program was an evening session addressed by Willis H. Carrier, who spoke on the future of air conditioning. . . . He felt there would be no great changes that would revolutionize present conceptions of air conditioning methods. . . . He stated that during 1936, manufacturers sold 43 million dollars worth of air conditioning equipment and the installed cost amounted to some 70 million dollars. He predicted that these figures would be increased by 50 to 100 per cent this year.

■ **Appointment** by the American Standards Association research committee on Grounding of a new technical subcommittee to investigate the effect of grounding electric wires on piping systems brings to a head a controversial question and may eventually lead to important changes in requirements of the National Electrical Code, according to Charles F. Meyerherm, secretary of the research committee, in an article in the April "Industrial Standardization." . . . An extensive program directed by the American Water Works Association and the Edison Electric Institute will determine the effect of grounding on pipes and pipe contents, safety, and efficiency.

■ **According** to D. W. McLenegan, of the General Electric Co., in a paper presented before the Montreal branch of the Engineering Institute of Canada and published in a recent issue of "The Engineering Journal," the cost of air conditioning "will be lowered somewhat as the designs of new buildings are modified to aid the installation, as those engaged in planning new structures become more accustomed to considering air conditioning as an integral part of a building, rather than as an appendage. In return, the advance of air conditioning is helping the architect and the builder to secure public realization of the advantages of better building materials. The author has seen many cases where insulation, better window frames, better wiring, and even structural improvements have been adopted after their advantages had been seen from the particular standpoint of air conditioning."

■ **From the Empress of Canada** at sea, the manuscript for Rush D. Touton's article on putting air conditioning plants in shape for summer operation, published this month, was sent. . . . When he started on his way to Sumatra recently, Mr. Touton stopped at our Chicago offices, said he was getting up some instructions to leave behind him covering maintenance and operation of air conditioning systems under his charge. . . . His article, which you will find well worth reading, is the result.

■ **The Eastern Osteopathic Association** recently adopted the following resolution:

Whereas, proper air conditioning is a definite public health measure,

Whereas, oxygen is vital to health and the continued recirculation of air without the addition of a fresh supply of oxygen is harmful,

Whereas, proper humidity and temperature control creates comfort,

Whereas, the improper forced circulation of conditioned air is a threat to health by creating ill-directed drafts,

Therefore be it Resolved, That we urge public health officials to redouble their efforts in inspecting all places where air conditioning is in use, especially theaters, concert halls and vehicles of transportation, to the end that this potentially valuable development may fully serve mankind.

■ **Holland's** new transatlantic liner "Nieuw Amsterdam" will carry the largest air conditioning system afloat, according to L. R. Boulware of Carrier Corp., designers of the 300 ton cooling system, larger by 75 tons than that of the "Queen Mary" and 114 tons larger than the "Normandie" equipment. . . . The system conditions the liner's three dining rooms, modern theater, and beauty and barber shops. . . . Termed Holland's ship of peace, the "Nieuw Amsterdam" has been built without any provision for future armaments, which is said to make her unique among modern merchantmen.

■ **The Dust Control** Equipment Association has announced the appointment of an engineering committee, one of whose principal duties will consist of the development of standards for dust control practice. The committee will also be available for consulation by other associations or groups interested in the preparation of codes, development of regulations, etc. . . . Officers of the association are: president, H. B. Loxterman, Blaw-Knox Co.; vice president, M. A. Eiben, Northern Blower Co.; executive secretary-treasurer, Arthur J. Tuscany. . . . Recognizing the function of the American Welding Society as a technical spokesman for the welding industry, the N.E.M.A. electric welding section recently voted to furnish financial support for the expansion program proposed by the A.W.S. . . . The matter of immediate importance is the coordination of tests of men, machines and filler metal to relieve some of the unnecessary burden which duplication of test requirements has caused.

■ **A conference** on the teaching of mechanical engineering is to be held at Cambridge, Mass., June 28-29, under the auspices of the mechanical engineering division of the Society for the Promotion of Engineering Education. . . . The Mellon Institute, Pittsburgh, is dedicating its new building, May 5 to 9, and among the internationally known scientists taking part in the program are four Nobel laureates.

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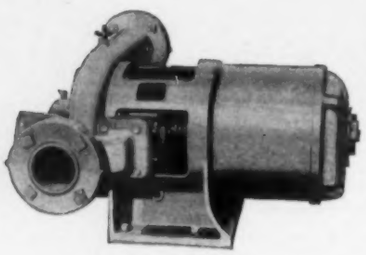

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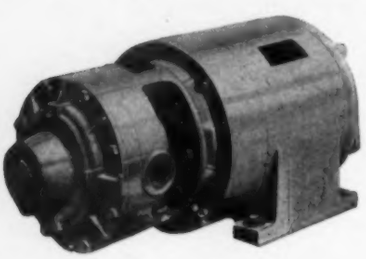

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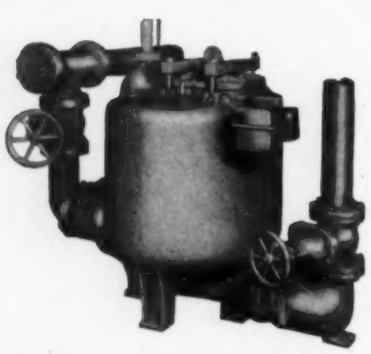

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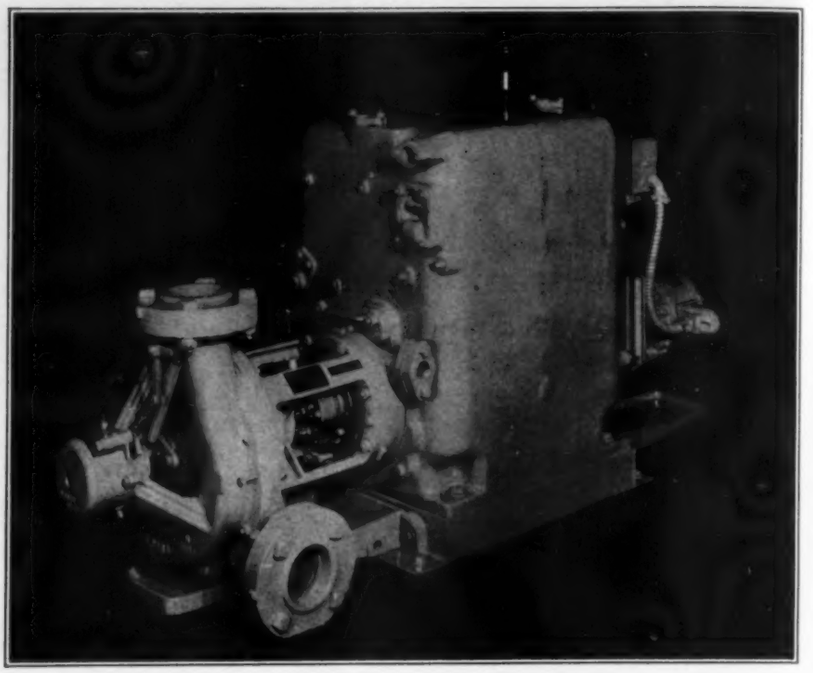

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Heating · Piping
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Vol. 9, No. 5 • *May, 1937*

Practical Standards for Air Conditioning Calculations

[Part 1]

By W. S. Bodinus*

THE purpose of this series of articles is to present to the beginning air conditioning engineer the practical procedure as well as standard tables and formulas for the various heat calculations. Naturally, the data available at the present time are not nearly as complete or authentic as we would like it to be; however, the information given here is the most recent which has been published, with the authority given after each for further and more complete analysis if desired by the reader.

Thermal calculations of all kinds (including those in heating, ventilating, and air conditioning work) must be carefully and completely made since the factors of safety (*i. e.* 10 to 30 per cent) generally used are very nominal, especially when compared with structural and mechanical design calculations where factors of safety from 300 to 800 per cent are used. The fear of air conditioning calculations by the average engineer was natural since there was a lack of experience as well as a lack of published data. Since most early air conditioning installations were made in industry, the doors of those plants equipped were closed to the knowledge seeker—not necessarily to keep the secrets of air conditioning but more often to hide a special manufacturing process. Only a few companies were engaged in the art of applying air conditioning equipment previous to 1930 and naturally the information regarding their procedure was not public property; neither were these companies free to disclose the processes of their clients.

Between 1920 and 1930 the comfort field began to develop, especially in theaters, where it proved to be a very profitable investment by opening the doors during summer. Until recently, there was almost a complete let-up in the industrial field but the number of comfort cooling installations climbed almost consistently. About this same time the engineering societies, manufacturers, and engineers began to release for general use data as well as reports of practical application. The several recent issues of the *A. S. H. V. E. Guide* have included most complete information.

In 1935 the Chicago comfort air conditioning standards committee, headed by Mr. John Howatt, presented a pamphlet giving standards and tables to be universally issued to the contractor and engineer; every engineer engaged in air conditioning should have a copy of this booklet since it gives a very complete opinion of the outstanding engineers, contractors and manufacturers for application to Chicago and vicinity.

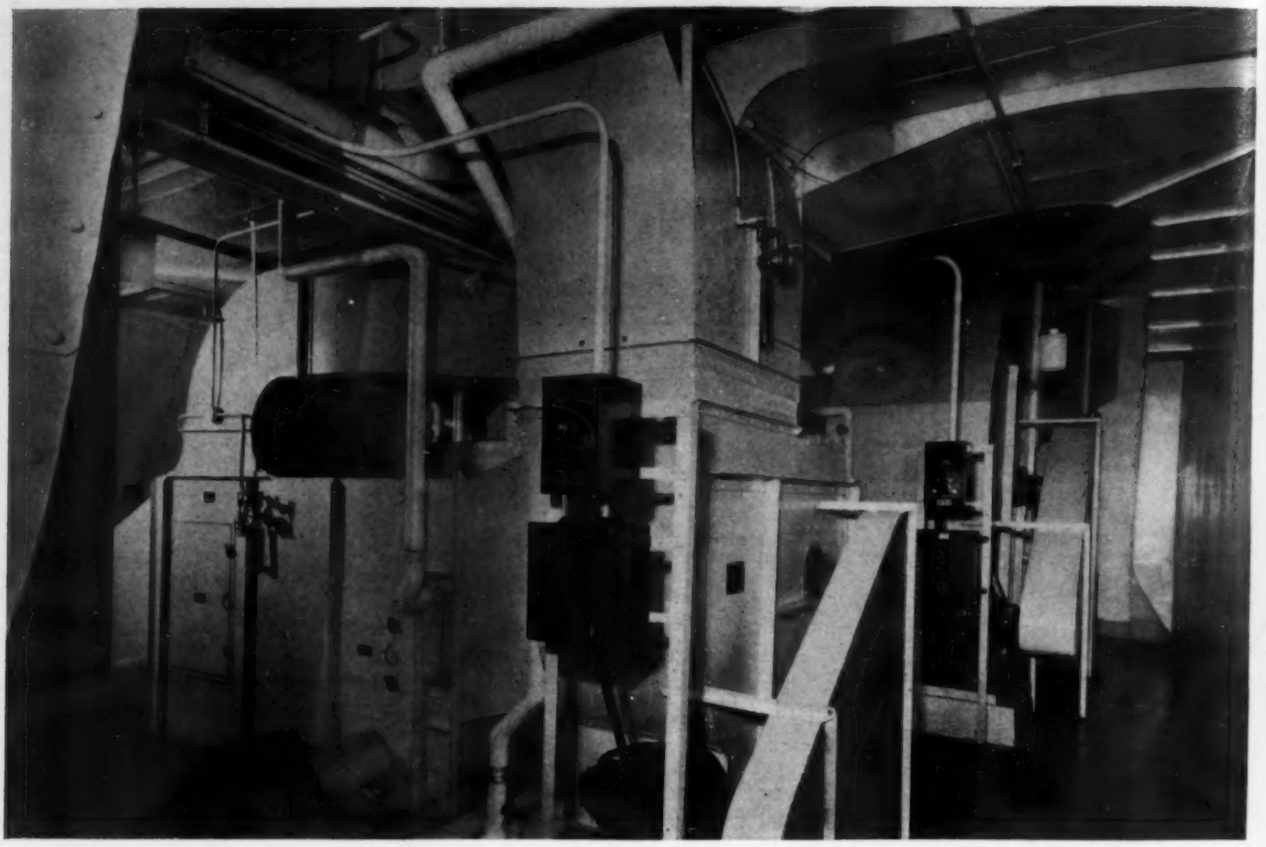
Comfort cooling for the small commercial installation will be covered in these articles and typical calculations will be made, removing as much as possible the mystery surrounding air conditioning. Larger installations will not be discussed since they present complex problems in total load, sun effect, zoning and people variation, that can only be analyzed by the experienced engineer. Industrial air conditioning problems likewise are problems for those thoroughly experienced in that kind of work, as well as those having a thorough knowledge of the business of that particular industrial application.

Review of the Basic Principles

Before proceeding with the heat estimate it probably will be in order to review the basic principles of psychrometry.

*Engineer. Carrier Corp.

A new series presenting for the beginning air conditioning engineer the practical procedures for making calculations, as well as standard tables and formulas for the various heat calculations . . . The author is particularly qualified to prepare this material, as he not only is actively engaged as an air conditioning engineer, but has also lectured to refrigeration and air conditioning classes



Air conditioning apparatus serving a bank building

DRY BULB TEMPERATURE is the temperature of the air as indicated on the ordinary thermometer. The dry bulb temperature also gives a measurement of the sensible heat generally listed in tables as the quantity of sensible heat above zero degrees. For instance, taking the basic formula

$$W \times (T_2 - T_1) \times C = \text{Btu}$$

W = weight of material in pounds
$T_2 - T_1$ = temperature differential
C = specific heat of substance
Btu = British thermal units, or quantity of heat.

If one pound of air is substituted in the formula and an average specific heat of 0.24 is used with a temperature rise from 0 to 70 F,

$$1 \text{ lb} \times (70 - 0) \times 0.24 = 16.8 \text{ Btu}$$

Thus the sensible heat present in air at 70 F from a level of 0 F is 16.8 Btu per lb.

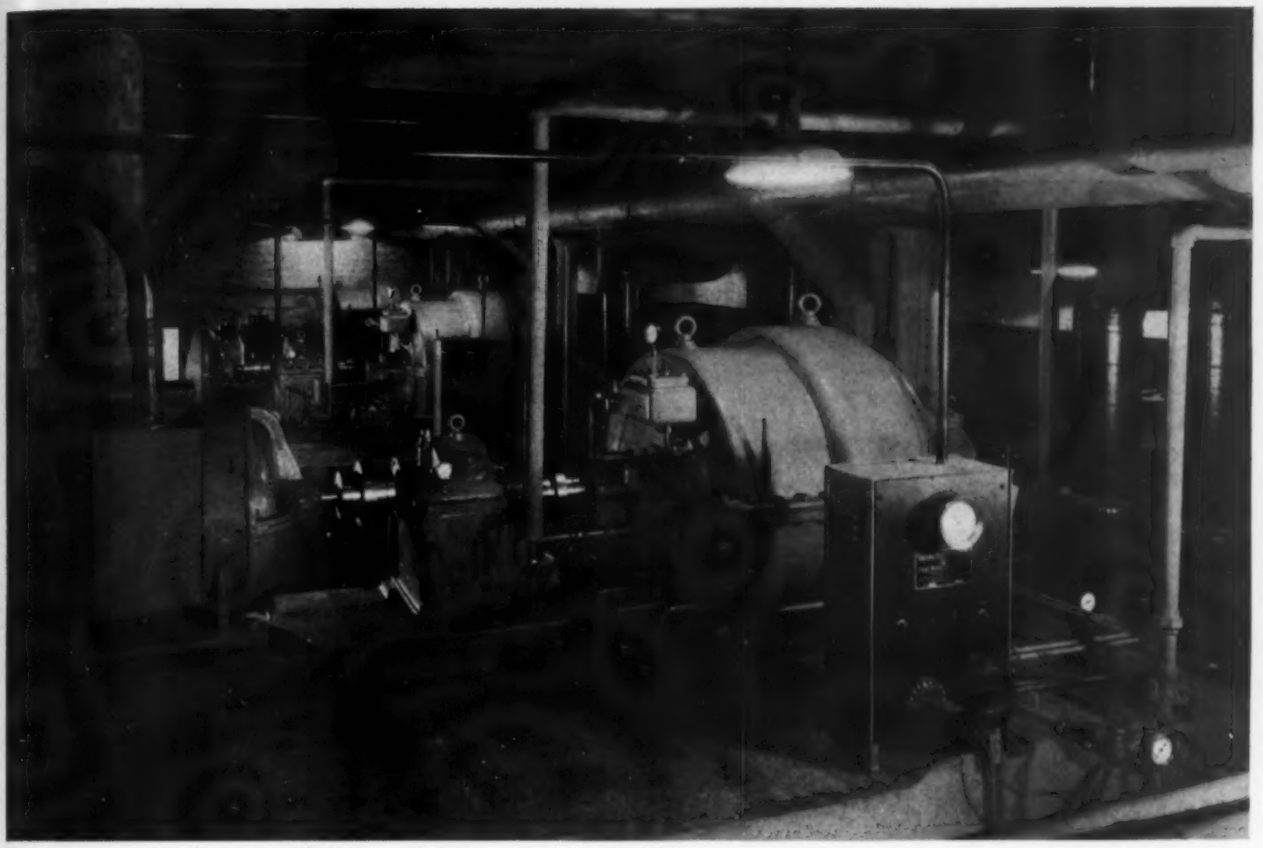
DEWPOINT TEMPERATURE is the temperature at which dew forms in the atmosphere; thus, when we reduce the temperature of ordinary atmosphere sufficiently, moisture (water vapor) begins to separate from the air in the form of mist or dew, because the air can no longer retain it. For example, when a cold glass of water is placed in ordinary atmosphere, dew forms on the outside of the glass. Thus, in an air conditioning apparatus one of the features of design is to provide a cooling coil of sufficient surface area as well as at a low enough temperature to precipitate the excess moisture. When a dehumidifier spray apparatus is used the water spray itself becomes the surface area as well as being at a temperature below the dewpoint to actually cause precipitation of moisture rather than addition of moisture. The dewpoint temperature also establishes the quantity of moisture in air and tables as well as psychrometric charts are made on the basis of grains of water vapor per pound of air. There are 7000 grains to one pound, so if a pound of air contains 100 grains of water it has 1/70th of a pound of water in it.

ABSOLUTE HUMIDITY—The quantity of water vapor in air given in grains per pound of air is called the absolute humidity. Since atmospheric air is a mixture of gases (of which water vapor is a part), the mixture exerts a total pressure (called atmospheric pressure, 14.696 lb per sq in) which is the sum of the partial pressures of each of the constituent gases. Thus the quantity of water vapor in the air is proportional to the pressure it exerts and is the same as if the water vapor existed alone. The pressure of the water vapor at dewpoint temperature of 67 F is 0.66638 in. of mercury absolute pressure. Then the atmospheric pressure of 14.696 lb absolute (equal to 29.92 in. of mercury absolute pressure) makes the balance of the mixture of gases in atmospheric air equal to (29.92—0.66638) or 29.25362 (approximately 14.3 lb) absolute. Since the water vapor pressure is 0.66638 we can determine the quantity in pounds or grains of water vapor from the following:

$$W = 4354 \left(\frac{e}{B - e} \right)$$

$$= 4354 \left(\frac{0.66638}{29.25362} \right)$$

= 99.0 grains of water vapor per pound of atmospheric air.



Centrifugal refrigeration machine for air conditioning an office building

RELATIVE HUMIDITY is the ratio of the amount of actual water vapor contained in the air to the total amount that would be present if the air were saturated at that temperature. Thus, if the relative humidity of certain air is said to be 50 per cent, then the air contains 50 per cent of the amount of water vapor that would be required for the complete saturation of the air at that particular temperature. When there is no addition or extraction of moisture, the relative humidity decreases with an increase in dry bulb temperature and the relative humidity increases with a decrease in dry bulb temperature. If the dry bulb is reduced sufficiently the relative humidity eventually becomes 100 per cent and the air is saturated. Further reduction in temperature then causes precipitation of moisture (dehumidification). Control of relative humidity is very important in air conditioning installations, both for the comfort of humans as well as industrial applications since many materials are hygroscopic, such as textiles, wood, rayon, food products, etc.

The following is an approximate formula for determining the relative humidity:

$$\frac{Wdp}{Wdb} = R.H.$$

$R.H.$ = Relative humidity
Wdp = grains of water vapor at dewpoint
Wdb = grains of water vapor at dry bulb

Thus with air at a given dry bulb temperature with a known dewpoint the approximate relative humidity can easily be determined. For instance, air at 80 F dry bulb and 60 F dewpoint would have a relative humidity as follows:

Grains of moisture per cubic foot at 60 F = 5.795
Grains of moisture per cubic foot at 80 F = 11.040

$$R.H. = \frac{Wdp}{Wdb} = \frac{5.795}{11.040} = 52 \text{ per cent}$$

The value 52 per cent is only approximately correct since the density of the cubic foot of air is different at the 60 F and 80 F conditions. Exactly, the relative humidity is the ratio of the partial pressures of the water vapor at the two temperatures.

The term *relative humidity* is used by many engineers in a careless way to mean "feeling or quality of dryness of the air" but, as a matter of fact, air with a relative humidity of 60 per cent when the dry bulb is 40 F is dryer than air with a relative humidity of 10 per cent when the dry bulb is 100 F. Relative humidity is the relationship of water vapor in the air at the dewpoint temperature to the amount that would be in the air if the air were saturated at the dry bulb; in other words, it is the relation between the water vapor pressure at the dewpoint to the water vapor pressure at the dry bulb (saturated condition). In reality it can be considered as a thermal condition of steam (or water vapor) at one pressure compared with steam (or water vapor) at a higher pressure. Since the flow of water vapor tends to be from the higher pressure to the lower pressure, drying is accomplished because a wet object tends to take the wet bulb temperature of the air (adiabatic saturation at the wet bulb) which in turn establishes the final water temperature and vapor pressure of the water in the object. This temperature is higher than the dew point

of the air and thus the greater the spread between the room wet bulb and the room dewpoint the greater the evaporating rate. The exact relative humidity can be obtained by dividing the vapor pressure corresponding to the dewpoint by the vapor pressure corresponding to the dry bulb and multiplying by 100.

WET BULB TEMPERATURE—The wet bulb temperature is read on an ordinary thermometer whose bulb has been covered with a wick that is saturated with water. The evaporating effect of the water thus reduces the dry bulb temperature to the so-called wet bulb, which always falls between the dry bulb temperature and dewpoint temperature. The wet bulb temperature is the temperature at which adiabatic[1] saturation takes place; that is, the dry bulb temperature is reduced (sensible heat) and the dewpoint temperature (latent heat) is increased, both meeting at the wet bulb. During this change there has been no addition or extraction of heat (adiabatic) since the dry bulb temperature is reduced at the expense of an increased dewpoint temperature. This was originally pointed out by W. H. Carrier in the *A. S. M. E. Transactions* for 1911. Mr. Carrier further established the fact that the wet bulb temperature represented the temperature which definitely indicated total heat of atmospheric air and the values were tabulated and graphically plotted against the wet bulb on his psychrometric chart. As stated before the wet bulb can be conveniently determined by applying a wetted wick to an ordinary thermometer.

The total heat above 0 F in a mixture of air at 80 F dry bulb, 60 F dewpoint and 67 F wet bulb, 52 per cent relative humidity, is

Sensible heat of air $= 0.24 \times (0$ to $67)$	$= 16.08$
Latent heat $= 0.01417 \times [1059.2 + (0.45 \times 67)]$	$= 15.43$
Sum	$= 31.51$

where 0.24 = Average specific heat of air.
67 = Wet bulb temperature.
0.01417 = Pounds of water vapor per pound of air.
0.45 = Average specific heat of water vapor.
1059.2 = A constant.

Thus the total heat per pound of air at 80 F dry bulb and 60 F dewpoint is 31.51 Btu per lb. The total heat could be determined also by taking the sensible heat of dry air, plus the latent heat of water vapor, plus the sensible heat of superheated water vapor from the dewpoint to the dry bulb, and plus the heat of liquid to the dewpoint temperature. All of these quantities would be taken above some base point, such as 0 F. This is quite a lengthy and difficult calculation which indicates the convenience of Mr. Carrier's wet bulb theory.

The principle of adiabatic saturation at the wet bulb is made use of by applying spray type dehumidifiers for air conditioning apparatus, so that intermediate season cooling during spring and fall can simply be accomplished by regulating dampers and control valves from a thermostat in the spray chamber. Also many days in mid-summer have an outside wet bulb temperature low enough to accomplish the cooling effect simply by using this principle with the necessary control dampers on the outside air and return air regulated by the dewpoint thermostat.

VOLUME OF AIR—The average weight of ordinary atmospheric air in summer is approximately 0.075 lb per cu ft, or inversely—13.5 cu ft of air equals 1 lb. This is determined from the basic formula $PV = WRT$

P = Pressure in lb per sq ft absolute
V = Volume in cu ft
W = Weight in lb
R = Gas constant, 53.34 for air
T = Absolute temperature, F

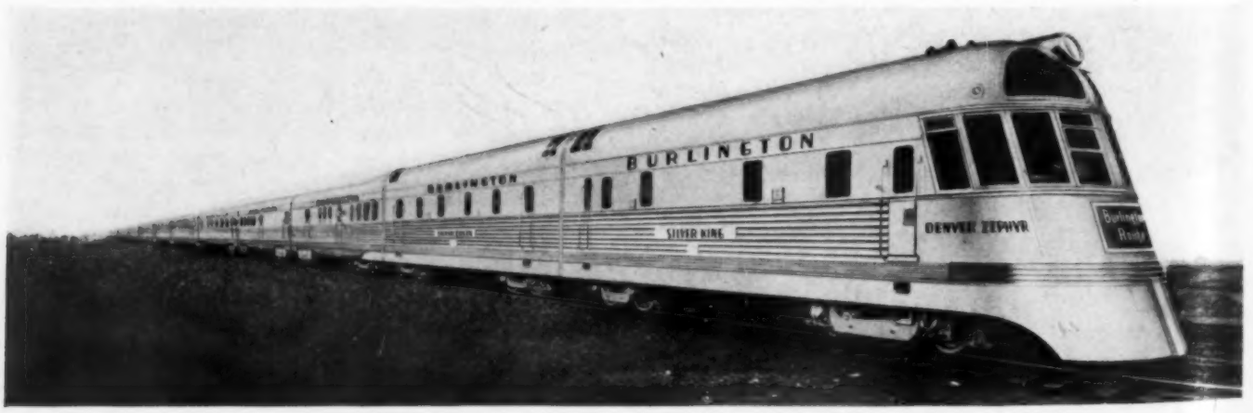
For one cubic foot of dry air at 70 F and atmospheric pressure at 14.696 lb per sq in. the weight W becomes

$$W = \frac{14.696 \times 144 \times 1}{53.34 \times 530} = 0.075 \text{ lb per cu ft}$$

This is the weight of dry air and since atmospheric air always contains a certain amount of moisture (water vapor) the weight of atmospheric air is lighter than dry air at the same temperature. For instance, one cubic foot of atmospheric air at 70 F saturated with moisture would weigh only 0.07425 lb.

In next month's article, Carrier's psychrometric chart will be shown, and examples explaining its use will be given.

[1]An adiabatic process is one in which there is no gain or loss of heat.



The Burlington's air conditioned "Denver Zephyr"

Practical Suggestions for

Improving Plant Piping Systems

Walter E. Hume* discusses the matter of better, more economical results from plant piping services, describes what may be done to reduce steam consumption

PUTTING plant piping systems in shape for better, more economical results requires first an understanding of how heat is transferred to the process, because—strange as it may seem—some confusion of ideas may often be found regarding steam by its users. This approach to the problem is of value in that the cooperation of steamfitters, operators and maintenance men is essential if, assuming the piping system itself is laid out properly, we are to get better, more economical results from its use.

How often we hear it said around the shop, "Steam at twice the pressure is twice as hot." A frequent cause of misunderstanding is the failure to distinguish between temperature and heat. Again, it is sometimes necessary to make clear that steam has weight, because pressure is also spoken of in terms of pounds. An informal monthly meeting of interested people when questions can be asked and problems discussed is one means of getting information out where it does some good.

Explanation of how processes are affected by wet and dry steam; clearing up of the idea about superheated steam being of particular value for heating purposes; information on the sizing of reducing valves; and, finally, the importance of active circulation of the heating medium should be given to the operating force.

Encourage suggestions. The knowledge that a suggestion made by a pipe fitter will reach the plant superintendent and be given consideration goes a long way toward getting cooperation in keeping a system in shape for better results.

Economy in Use of Steam

The development of modern appliances to secure higher efficiencies in the boiler plant has made possible effective cost control insofar as the *generation* of steam is concerned. It is also in the *use* of steam by process units that opportunities may be found to reduce the cost of service charges. However high the efficiency of control in the generating plant may be, the demand obviously depends upon the amount of steam used. Steam that is wasted and that which is effectively used make up the total consumption. The point to be emphasized is that waste of steam in its transmission and its use at the point of service are often far greater than wastes in the boiler plant.

We may establish control of any process, set up standards of performance for operators, check and guard against waste motion, provide rest periods to reduce fatigue and at the same time may overlook waste in the use of steam, air and water constantly occurring coincident with a highly developed system of process control. The steam that leaks never gets tired of so doing.

Inspection of Piping

Suppose we make a trip through a boiler plant and notice how the mains and headers are insulated. We shall, no doubt, find double thick covering over pipes and possibly a canvas jacket sewed on. Let us continue our inspection throughout the plant where smaller size piping serves process units or carries steam to the heating system. It is here, we are likely to find, that much less consideration has been given and care taken to save steam. The fact that pipes are smaller seems to influence the degree of care taken.

Less efficient insulation may often be used for small pipes. As a matter of fact it is more economical to use the best insulation available because a poor grade requires a thickness out of proportion to the size of the pipe and in the aggregate the amount of small pipe surface requiring insulation is often greater than that of the mains in the boiler house. Over here, we find that the ends of the 3 ft lengths of sectional covering on a 4 in. pipe have not been properly lapped. The covering is loose and there is an open joint between each length of covering; a simple item, but responsible for a waste of heat. It may cost $0.50 a month to leave a 1 ft length of 4 in. pipe uncovered and it doesn't require many open joints to equal a 12 in. length of uncovered pipe.

Cost of Leaks

Most of us are familiar with tables that show what it costs to allow steam to "leak out." We find that for a given pressure, a $\frac{1}{32}$ in. diameter opening will waste $1.50 per month when steam costs $0.50 per 1000 lb. If the opening be ¼ in. the waste would amount to $101.50. Now, a ¼ in. leak would not be allowed to continue for very long but a $\frac{1}{32}$ in. leak from which steam more often than not escapes unnoticed, frequently continues for a long period of time. It is not difficult to understand that with several thousand joints and valves a hundred or more small leaks may usually be found.

If it be compressed air that is leaking, a ⅛ in. diameter opening might waste $55.73 per month and the operator who leaves an air hose on while he goes out to lunch might waste $0.077.

For every hour that cooling water through a ⅜ in. diameter pipe is left running needlessly it may cost $0.048.

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How can such wastes be detected and stopped? There are two of our senses that may be put to work: hearing and seeing. Unfortunately the first is often nullified because of noise from surrounding machinery and the second voided because the leak is hidden. "You have to listen and look and then look and listen"; in other words become *steam conscious* in the matter of waste.

Cooperation, posting of tables showing cost of wasted steam, air and water, together with a campaign of education, will help, but the efforts must be unfailing. Spasmodic attempts at waste reduction carry their own failure.

Let it be stated here that increased cost from waste is preventable; in this connection there's an old, old saying about the ounce of prevention that covers the problem of steam waste or waste of any other kind.

Inefficient Heat Transfer

Inefficient heat transfer represents waste. Water-logging or the accumulation of condensate is one cause and air locking or accumulation of air in the system is another. Failure to free a system of air is equivalent to setting up a very good insulation in the wrong place.

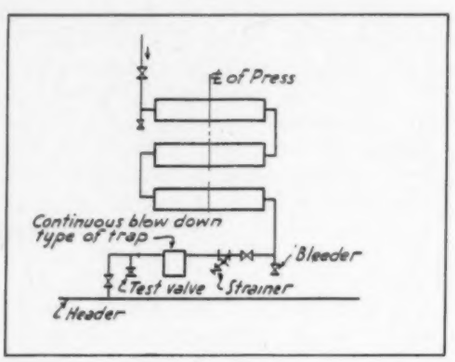


Fig. 1

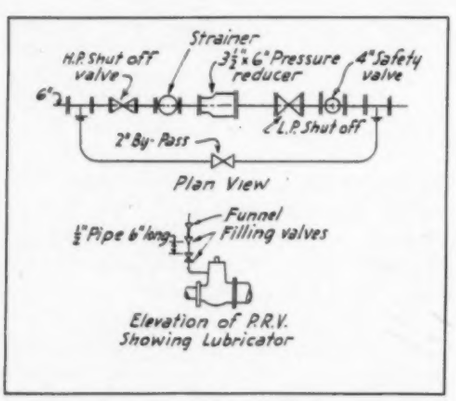


Fig. 2

Failure to drip a main at suitable locations, particularly the end of a line, causes water-logging. We know what it means when a highway sign reads, "Dead End Street." The same thing applies to steam at the end of a line. It is "No Thoroughfare" in each case, but with steam this condition means plugging the line with water.

A characteristic of steam, never to be forgotten, is its tendency to return to its native state, water. We expend work on the water and change its state, but in its changed condition, it never loses an opportunity to revert back. In so doing, it gives up the work expended upon it. If in so doing, it does useful work, well and good. Otherwise, only wasted effort is obtained for the expenditure of work put into the water when generating steam.

More effective heat transfer from steam to product reduces cost, active circulation in whatever form applied being the first essential. In paper drying, curing rubber, heating oil tanks, warming tables, presses, jacketed vessels, or in whatever form the heating process takes place, effective heat transfer reduces cost.

Use of Exhaust Steam

One of the methods by which savings may be made is by the utilization of exhaust steam which ordinarily may be used to heat the feedwater for the boiler. In some cases, it is used for processes requiring hot water.

In our Butler plant steam is purchased from the outside. Not having the direct use of exhaust steam for process purposes made it necessary to devise means to utilize as much of it as possible. We therefore connected trap discharges to soaking tanks, warming tables, drying bins and heating coils in certain departments located on floors below the trap discharge, in all cases replacing the use of live steam.

We have more than 100 warming tables and tests showed that 1½ lb of steam per sq ft per hr is the average use, making a cost of over $4,000 per year for steam. Steam at 80 lb pressure was formerly used. After a series of tests it was found that 30 lb pressure resulted in a temperature ample to keep articles laid on the tables hot. In other cases, trap discharges bleeding the mains have been connected to the tables as a source of heat.

Operation of Auxiliary Equipment

However much we may endeavor to clear up misunderstandings regarding steam and how it serves process units, no matter how carefully we may plan and lay out the piping, the proper operation of what may be termed auxiliary equipment is largely responsible for the economical operation of the units served.

First in order is the use of suitable valves, properly located and kept in condition through careful inspection and maintenance. Gate valves should be used for shut-off purposes. Globe valves should be used for throttling or regulating the flow of steam. Frequent inspection and renewal of seats is essential to economical operation.

When we come to adoption of more efficient ways of utilizing steam for process, it is necessary to consider the three methods commonly used: (1) Open discharge of live steam. (2) Condensate discharge by means of a trap. (3) Recirculation of steam.

The first, open discharge of live steam, is often found in service and gives very satisfactory results because of rapid steam movement by using an open exhaust. The results, however, are obtained at a price. The waste is obvious. Attempts made to reduce the steam consumption by throttling the supply valve are not satisfactory because variation of the line pressure requires resetting of valves. When used on a battery of units, like presses

with variable temperatures, a common exhaust header is not practical where temperature requirements of presses are variable.

Our first attempt to cut steam waste on presses led to a series of tests with various types of traps. Difficulty was experienced in obtaining uniform temperatures. A common discharge header was unsuitable for obvious reasons. Providing an open discharge in the room from each press brought troubles of a different nature. The larger the trap the less satisfactory were results from the standpoint of uniform heat transfer. A small trap operating frequently points the way to a solution.

We have used with satisfaction traps of the continuous blow-down type for process units, as active movement of the steam throughout the system aids the maintenance of uniform temperatures. Among the continuous discharge types may be mentioned one in which there is a pulsating discharge of the condensate through an adjustable orifice. Such an arrangement permits a continuous flow of the condensate with a minimum discharge of live steam.

Thirty presses have been furnished with this type of equipment and give good results. Its principle limitation is an inability to handle the condensate quickly after a shut-down due to lack of pressure when steam is first turned on to a cold press. Fig. 1 shows a typical hookup to a press. Note that a common exhaust header is used but each exhaust line is furnished with a blow-down valve, strainer and test valve. This continuous discharge is used in conjunction with an automatic temperature regulator in the steam supply to each press.

The third arrangement refers to the recirculation of steam using an injector to induce return of the exhaust steam. This arrangement has been applied to a battery of comb presses and obtains the benefits of active circulation with a minimum loss of live steam. In addition to recirculation the combination supply and injector valve has been equipped with an automatic temperature control. The arrangement has, from the statement of the foreman in the department, given satisfaction from the standpoint of steam saving, uniform temperature and freedom from repairs.

In vulcanizing processes, the requirements are, generally, uniformity of temperature and dry steam. A series of experiments brought out some facts regarding temperature lag caused by failure to get rid of the air that accumulates in the ends of the vulcanizers. The use of a top vent pipe, open at the beginning of the heat, remedied this trouble. Experiments with an injector and recirculation gave data on the benefit of active induced circulation and reduced steam consumption not obtainable with an open exhaust system.

Where temperature uniformity is an essential, tests made with an automatic temperature control brought out clearly the necessity for dry steam. To eliminate excessive moisture is essential for proper operation and the setting of the exhaust valve in an open exhaust system is of much importance. The success of an automatic temperature control as to its ability to maintain temperature and pressure without variation is governed largely by the manipulation of the exhaust valve from the chamber. As an example, in one instance, an automatic temperature control was installed for test. The operator reported that the control would not hold the pressure. Investigation showed that the waste valve was open too wide. A reduction was made in the opening of this valve and the control functioned perfectly.

Installation of Steam Lines

It is scarcely necessary to mention that a steam main supplying a number of circuits should, for economy in installation cost, be reduced as the take-off branches are reached. To insure dry steam, branch lines should be taken off at the top of the main. Each branch should have a shut-off valve at point of take-off. Is it unnecessary to make the foregoing statement? The answer is, "Look around the average plant and see where, for 'false economy,' valves have been omitted."

It is inadvisable to install mains which are greatly in excess of requirements. A slow rate of flow results in condensation and wet steam. On the other hand, where steam is purchased from a separate plant located a considerable distance away, care must be taken to so schedule the starting up of various units that a heavy overload demand on the plant does not occur. This is necessary for two reasons:

1. With certain types of boilers the reserve capacity may be limited and an unusually heavy drain will have a tendency to cause water to be poured over into the mains; therefore, a careful schedule must be adhered to where there are a number of process units requiring steam in large quantities when they are started up.
2. An excessive pressure drop in the mains caused by overload demand may very easily upset the heat balances required for various processes.

Provision for Expansion

A survey made in a textile plant disclosed an abnormal amount of steam consumption over week-ends and when it was asked why a certain line was not shut off, the answer was "It pulled apart when it cooled down." The particular line in question was approximately 300 ft long, carrying steam at 100 lb pressure; it had no provision for expansion and was without an anchor. Quite obviously the remedy was to alter the line and install an expansion bend. Thereafter no trouble was experienced from shutting down the line during periods of non-production.

Failure to provide for expansion means high maintenance costs.

Reducing Valves

Where process conditions require pressure lower than that supplied by the mains, the use of pressure reducers becomes necessary. If possible, reducers should be grouped to suit pressure requirements and too much emphasis cannot be placed upon the proper size of valve selected. More trouble is experienced from oversize reducing valves than from any other single source. This condition is made worse when week-end and holiday steam consumption is at a minimum. Chattering, wire-drawing and building up of excess pressure occurs.

In selecting a reducing valve, we have found that a type wherein the discharge side has been increased in area gives satisfactory service. In other words, the increased area on the low pressure side is an integral part of the valve.

One other trouble of somewhat frequent occurrence is sticking of the valve and this has been overcome by providing a lubricator as indicated by Fig. 2.

Of equal importance is the installation of a strainer, bypass, safety valve and pressure gages.

Insulation

Lack of attention to proper erection of pipe insulation is responsible for a large amount of waste heat. It is very seldom that uninsulated pipes are allowed to remain in that condition for very long, but it is surprising to find how often temporary connections are left uncovered in the rush to get the steam "hooked up." Then the temporary lines become permanent and before we know it, the 1 in. line has been left uncovered for a month or more.

Standard insulation is supplied in 3 ft lengths with a coarse jacket attached which is provided slightly longer than the insulation so as to give a lap. More frequently than otherwise the insulation is installed "as is," which leaves a gap between each length of insulation.

Pipe insulation which is liable to be broken by trucks should be covered with a protective jacket or roofing felt.

Small pipes, particularly, should have the best grade of insulation available in order to keep the required thickness within reasonable limits. A check-up of the amount of heat lost at the surface of pipes would probably lead to an increase of insulation on the smaller sizes.

What May Be Accomplished

We have discussed ways for obtaining better and more economical operation of plant piping, stressing, at the beginning, the need for cooperation and understanding. How may results be brought home to those who are asked to reduce waste? It doesn't mean much to say that the steam costs $10,000 a month or that we use 15,000,000 lb of it. Such statements are not tied in to production activity. It is much better to establish a relation between goods produced and pounds of steam required to produce them. At our Butler plant we have set up a standard of 30 lb of steam per lb of compound mixed. Four years ago, the average was 70 lb of steam. Since the adoption of this standard, the average has dropped steadily until today we are within striking distance of the mark. This reduction shows what has been done in bringing about better, more economical results.

Whatever the product may be requiring steam, air and water, the question of cost reduction is subject to the same scrutiny, thought and action exercised in the control of process operations. It should not be difficult to find a working base from which a ratio may be determined to the end that a comparison of actual consumption with the standard laid down will indicate how efficiently the control of the service charge is being administered. The use of a control curve does not of itself insure a reduction in waste. To meet the standard laid down requires the active cooperation of each and every employee in the plant. In some respects, the activity of the industrial engineer has confined itself to the cost reduction of process operations. Is it not of relatively equal importance for the plant engineer to obtain cost reduction of the service rendered to the process operation by the use of steam, air and water?

Sterilizing Air

In a recent talk on inventions and our standard of living, A. W. Robertson of the Westinghouse Electric & Mfg. Co. prophesied that the future would bring ways and means of purifying the air we breathe in order to combat the contamination resulting from concentrating many people within the limits of our cities.

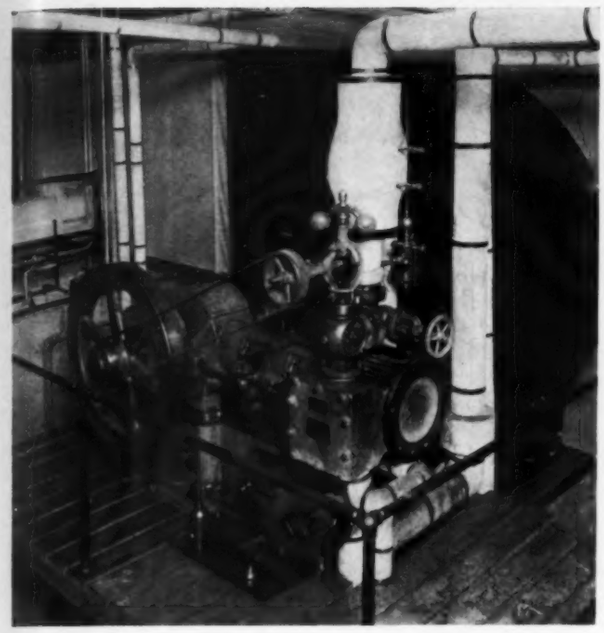
The sterilizing lamp is well adapted to do this work. Some very interesting results have been obtained with an experimental installation at Duke University hospital under the supervision of Dr. Deryl Hart who has believed for a long time that most of the infections of surgical cases were due to the organisms carried in the air and settling in open wounds during operations. Exhalations of the operating staff during the winter months when colds are more prevalent resulted in greater danger of infection of patients so that certain extensive operations such as thorcoplasty could not be successfully performed except in summer. After installing a group of eight sterilizing lamps on a framework surrounding the lighting unit over the operating table there was marked improvement in the results. Post-operative shock was reduced or eliminated, there was little temperature rise and this for a shorter period, and average time of healing was reduced from 21 days to 9 days. The operative staff was protected from the radiations by gowns, goggles and masks. Tests made on animals showed that radiation of open wounds for periods of 30 to 90 minutes resulted in no damage but healing appeared to be more rapid.

Petri plates sprayed lightly with a filtered suspension of organisms were rendered sterile in one minute while heavily sprayed plates required a longer time, there being one colony remaining after 15 minutes exposure. In the extreme corners of the room 13 ft from the group of sterilizing units 80 to 90 per cent of the floating organisms were killed while in the region of the patient, the instruments and dressings the bactericidal action was practically complete.

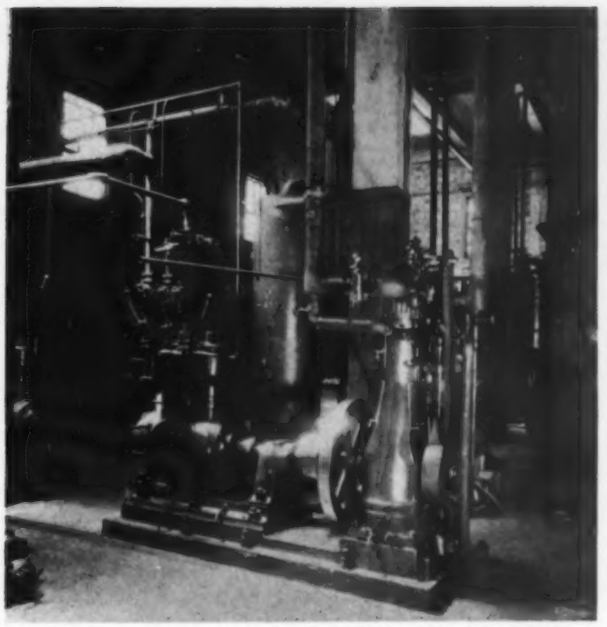
Radiation of the air entering the room even though all organisms were killed must be supplemented by direct radiation of the operative area to kill the organisms exhaled by persons in the room or brought in when they entered. Undoubtedly the hazards of surgical operations may be materially reduced by the air sterilization outlined and this may become one of the standardized hospital procedures.

Perhaps the efficiency of office workers may be increased by having the air supplied not only heated or cooled and dehumidified to meet health requirements but also treated to reduce the bacteria count, thus releasing additional energy for the task at hand instead of battling our unseen microscopic enemies.—A. R. DENNINGTON.*

*Westinghouse Lamp Co., Bloomfield, N. J. From a paper presented before the Toronto, Canada, section of the *A.I.E.E.*



Horizontal engine driving school ventilating fan



Engine driving refrigeration compressor at hotel

Making Steam Do Double Duty

F. J. Vonachen* Discusses Use of the Modern Steam Engine in Heating and Air Conditioning

IT it possible in many instances to improve plant heat balance and reduce operating cost by making the same steam do double duty, furnishing power for fans and refrigeration, and producing heat. Steam can be made to do double duty by passing it through a prime mover and using the exhaust for heating, or for steam jet refrigeration for air conditioning. That type of prime mover should be selected which will give the required power and at the same time an amount of exhaust steam that will not be excessive for the load.

The steam engine frequently best meets the requirements (1) because the steam rate remains low throughout the entire speed range, and even at the high back pressures which are sometimes necessary to obtain high temperature in the heating steam, so that the amount of exhaust best meets the requirements for heating and a good heat balance results; and (2) because it is a simple machine, requires little maintenance, and is readily operated. About 90 per cent of the heat originally in the steam is available in the exhaust for heating, so the power developed is a low cost byproduct.

Since some installations may require steam free from cylinder oil, it is well to mention that many steam engines are being operated with saturated steam without oil cylinder lubrication. With the cylinder properly constructed and glazed by the use of beeswax and graphite, it may be operated without any lubrication except possibly the introduction of a slight amount of beeswax and graphite at infrequent intervals. A solution of colloidal graphite in distilled water, introduced into the cylinder by a special mechanical cylinder lubricator, has also been used with success. The graphite concentration is low so that it does not interfere even if the steam comes in direct contact with the material or water to be heated, and its effect on boiler operation is considered advantageous.

Steam conditions offer no particular problem as steam engines are suitable for a wide range of conditions—from 20 to 400 lb pressure, from very wet steam to 750 F total temperature, from vacuum to 75 lb and higher back pressure. There is no limit to the amount of back pressure, if there is sufficient initial steam pressure to produce the required power within the limits of practical cylinder bores.

Cost of generating this power consists of operating and fixed costs. Operating costs are lubrication cost and about 10 per cent of the steam cost (the other 90 per cent of the steam cost being charged to heating.) Fixed costs based on a 20 year life are interest on investment, say six per cent; annual depreciation, five per cent; annual maintenance, two per cent. As a general rule, no extra operating labor is necessary.

While it is difficult to prepare byproduct power cost charts for the many conditions existing in various plants, it might be well to give a range of expected costs for both large and small engines for one particular set of average conditions. Assume the following: 150 lb ga steam pressure, no superheat, 5 lb ga back pressure, 2000 hr operation per year with an average load 50 per cent of full load, all exhaust steam used, and steam costing

*Troy Engine & Machine Co.

50c per 1000 lb. Taking into consideration all operating and fixed costs, experience indicates the cost of the byproduct power would range from 0.75c per bhp per hr (1c per kwhr) in a small 12 hp engine, to 0.3c per bhp per hr (0.4c per kwhr) in a larger 225 hp engine. These figures are based on an average load which is 50 per cent of full load; with higher average load, the cost figures would be still lower.

The following survey dealing with an 18 in. bore × 12 in. stroke steam engine driving a ventilating fan at 200 rpm in a school is pertinent. The engine receives steam at 30 lb ga and exhausts to the heating mains, at 2 lb ga, 90 per cent of the heat originally available in the live steam. The school accommodates 855 pupils in 19 class rooms. The fan driven delivers 69,300 cfm of air against 1½ in. static pressure. The engine is operated eight hours daily, 20 days per month, 10 months per year.

Dry, saturated steam at 30 lb ga contains 1,171.6 Btu per lb. The normal fan load on the engine is 36 bhp. Tests have shown that the engine requires 34 lb of steam per bhp delivered. The normal hourly steam consumption of the engine thus is 1,224 lb. Calculations which take into consideration the mechanical load on the engine, engine efficiency and the thermal equivalent of work done, show that the exhaust steam contains 1,069.4 Btu per lb. Considering only the effective heat of the steam (above 185 F feedwater temperature), 90 per cent of the heat originally available in the steam is exhausted into the heating mains. On this basis, the engine is properly charged with but 10 per cent of the total cost of steam consumed.

The following tabulation shows the cost of operating the engine:

First cost of engine	$2200.00
Anticipated useful life, years	20
Annual service, hours	1600
Annual Engine Costs	
Depreciation	$110.00
Average interest (at 6 per cent)	69.30
Maintenance and repair (2 per cent allowance)	44.00
Lubrication (80 gal estimated)	64.00
Steam (1,224 lb per hr × 1600 hr at $0.49 per M lb × 0.10	95.96
Total annual engine cost	$383.26
Total daily engine cost (200 days per year)	$ 1.92

It is estimated this engine is paying for itself out of savings every three years.

There are many types of layouts that may be used. One interesting installation is an engine driving a compressor in a hotel for general refrigeration, the exhaust being used in a jet refrigeration system for air conditioning in the summer, and for heating purposes in the winter so that byproduct power is obtained throughout the year.

Turbines for Heat Balance Purposes

By S. O. Maxwell*

I WOULD like to outline briefly some of the possibilities of obtaining economical byproduct power through the proper selection and design of steam turbines for heat balance purposes as drives for centrifugal refrigeration machines.

The centrifugal refrigeration machine operates at speeds closely paralleling the most efficient speeds of turbines, which means that it is ideally suited for direct connection to a turbine, which in turn, presents the possibility of applying the complete unit to a number of different heat balance propositions. This is especially true when all, or a large part, of the exhaust steam from the turbine can be used for process or heating, power therefore being made a byproduct of the process steam. It is also true, in some cases, where there is no use for the exhaust steam, and where high pressure steam is available, and the necessary power can be generated in a straight condensing turbine, that cheaper than the equivalent electrical power can be generated or purchased.

At one brewery, there is a simple non-condensing turbine, taking steam directly from the boilers, exhausting it at low pressure to the brewery, for use in cooking and heating; therefore, developing all the required power to drive the machine as a byproduct of the heating steam, as the turbine extracts only a small percentage of the heat in the steam in generating the power.

The machines installed at another brewery are an example of the use of a bleeder turbine where steam is taken at high pressure direct from the boilers to the turbine, and is bled out at a lower pressure in varying quantities for brewery use, and such additional steam that may be required to carry the load is then passed through to a condenser. Here again, the power produced by that portion of the steam which is bled out to the brewery is obtained at practically no cost.

The machines installed at Madison Square Garden, New York, illustrate their flexible installation, where the turbines are operated non-condensing in the winter time, the exhaust steam being used for building heating, the power at that time being obtained again as a byproduct of the heating steam. In the summer, the same turbines are operated condensing, when there is no use for the exhaust.

The R. H. Macy machines are typical of the low pressure type of turbines, where the turbines operate on exhaust steam from other equipment which was formerly wasted; these turbines, at times, actually operating from one vacuum to another.

The installation at Gimbel's department store consists of straight condensing turbines; in this installation it developed that power generated from this type of turbine is cheaper than the purchase of the equivalent electrical power.

There is also an installation in one of the large office buildings in New York, where it is cheaper to purchase steam from the utility and put it through a condensing turbine, than it is to purchase power and put it through a motor.

These installations cover about every possible application, and these in turn, can be applied to any industry or plant which has steam available, and particularly to those which have a demand for, or an excess of, low pressure steam.

*Terry Steam Turbine Co. From a recent address before the *A. S. R. E.* at Newark, N. J.

Rush D. Touton,* Authority on Industrial Air Conditioning, Suggests What to Do Now to

Put Your Air Conditioning Plant in Shape for Summer Operation

THE summer months represent the period of maximum load on air conditioning plants, and for this reason it is important that such equipment be made ready now for efficient operation. Whenever possible, this work is performed during the winter at the plants of Bayuk Cigars, Inc., makers of the "Phillie" cigar. Due to the constant conditioning requirement for the many millions of pounds of tobacco in process, it is sometimes impracticable to interrupt operation during low dewpoint weather, and a pre-season checking is made a part of the spring and summer air conditioning routine. The basic program follows the outline given in the succeeding paragraphs, and the observations included are the result of well over ten years' experience with industrial air conditioning.

Inspect the Structure

The structure or enclosure is first to be considered, bearing in mind the high cost of heat extraction. Any cracks or other failures which have developed during cold weather, particularly in the insulation, must be repaired. Obviously, any supposedly insulated wall or other surface which showed evidence of condensation during the winter is in no condition to prevent the inward flow of heat, and in any event, such ineffectiveness must be corrected before next fall. Shades or venetian blinds cut down the inflow of undesirable heat through glass areas.

Air locks or vestibules can well be employed where the differential between outside and inside conditions warrants them. In one instance such a measure was entirely responsible for converting a dubious cooling system into a satisfactory one.

The Air Distribution System

For summer air conditioning, a relatively high rate of air circulation is desirable. Where cooling is not provided, an increase in the rate of air motion is a useful means of lowering the effective temperature, and will make for better comfort for the workers and lessened regain in some hygroscopic materials. With cooling equipment incorporated in the system, it is advantageous to circulate large amounts of air at temperatures not greatly lower than the desired room conditions, thus minimizing the possibility of drafts and permitting the cooling medium temperature to be carried at a high and economical level.

In many cases, it will be easily possible to increase the fan capacity, and preserve the lower air volume arrangement for flexibility in conveniently meeting cold weather conditions. Such a set-up will readily demonstrate its merit.

*Technical Director, Bayuk Cigars, Inc. Member of Board of Consulting and Contributing Editors.

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As the coming months represent the period of maximum load on industrial and commercial air conditioning plants, it is important that the equipment be in A-1 shape to do the job. The suggestions for inspection and maintenance given here are based on over ten years' experience with systems handling more than 250,000 cfm of air in the tobacco industry

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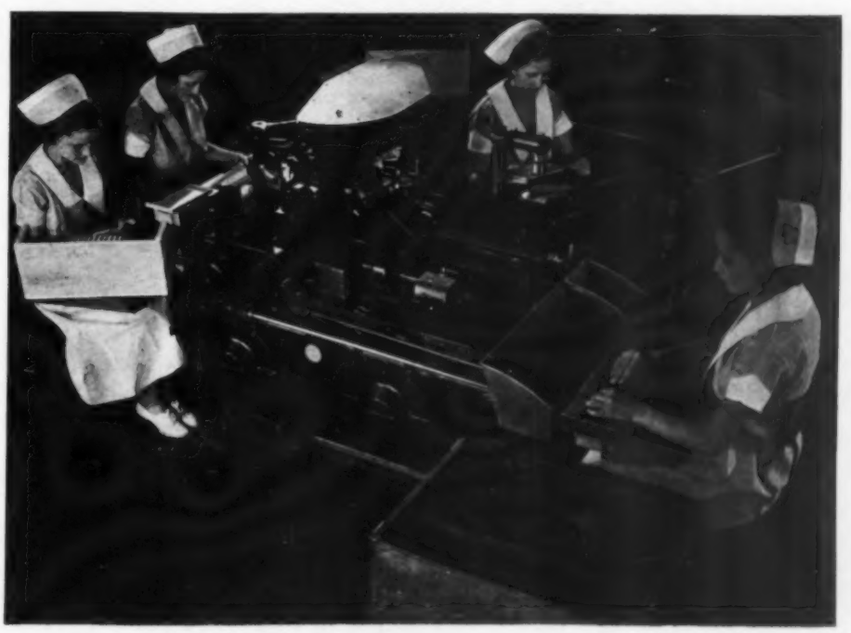


A close examination should be made of the duct system—especially when high dewpoint air has been carried through colder areas, causing condensation and consequent corrosion. The rust must be removed, and if the insulating material is damaged or inadequate, both it and the duct surface should be repaired.

Many systems are provided with air treating equipment which functions well, but due to the long transmission through bare or poorly insulated ducts, the air at the point of use is ineffective. The wet bulb depression should be carefully checked at the air outlets, and if found to have changed considerably over the figure at the fan outlet, steps should be taken to correct this loss.

It may be found that the duct system, the filters, or the fins on the indirect heating surface have collected foreign material to a sufficient degree to restrict the air passage. The air volume delivered may be large enough for the lesser loads, but will certainly be inadequate for peak demands. The system should be examined and cleaned from the inside if possible. It is advisable to perform the cleaning on week ends, so that the dirt blown from the outlets may be collected before the product or equipment is damaged. In any case, covers should be placed as required before this operation is begun.

The diffusers or outlets through which the conditioned air is finally delivered should be inspected and frequently adjusted to render the best summer service. Deflectors may be placed in duct or outlet systems to direct the maximum air quantity to the desired location, and in some cases the cooled air must be delivered in a manner quite distinct from the heated air for best results and to avoid drafts. In one instance, it was possible and distinctly practical to equip the overhead ducts with both side and bottom outlets to enable the air to be directed as required by seasonal conditions.

Again, the deflectors might be made easily adjustable to permit zoning as dictated by changing sun effect during the early and late hours of the day.

Test the Refrigeration Plant

The refrigeration assemblies deserve most careful attention—generally such equipment will not have been in use for some months, and must be thoroughly tested. This servicing is most conveniently and economically performed during the winter months, and if for any reason it has been delayed, it should now be completed. This will include an inspection of the electrical or other driving arrangement, and it is advisable to make certain that the use of this power, since it often includes much starting and stopping, will not cause other motors, lights, etc., to operate irregularly, and possibly cause tripping of the main circuit breakers.

Both the high and low sides of the system must be tight, and a compressed air or similar pressure test may at this season of the year be carried on for a sufficiently long period to locate the small and often neglected leaks which are so disconcerting later.

The condition of the valves, oiling system and the pistons and cylinders should be observed thoroughly. Scoring, pitting, and erratic action in these parts can only become more serious with additional operation. If any evidence of improper lubrication is discovered, it will be advisable to consult the makers for recommendations. The small number of hours the unit is operated each year makes it highly inadvisable to attempt to economize in initial oil costs.

At this time, it may be possible to correct disturbing noise conditions originating in the compressor, by installing isolating dampers in the foundation arrangement. Such devices are inexpensive, and where the unit is located close to an occupied space, the results from this treatment are well worth while.

All gages should be calibrated accurately. If not found responsive, the defective units can well be replaced, at a considerable saving in operating cost.

The condensing arrangement is of great importance to successful operation of the refrigeration. Furthermore, with water cooled types the greatest load usually develops at a time when the source of supply is being subjected to high demands for other purposes, resulting in scarcity. In any event, a 10 per cent loss of efficiency may well creep in without being noticed if the head pressure is allowed to rise beyond the design figure.

All heat transfer surfaces must be thoroughly cleaned. For the more corrosive waters improved results may be obtained by replacing steel tubes with copper lined or other tubes which are not affected by the chemical nature of the water used. Aside from eliminating frequent replacement, such tubing will remain clear and thus maintain a high rate of heat flow from gas to water throughout the operating season.

It is impossible to lay too much stress on the proper design and functioning of the evaporator or low side of the cooling system. With other conditions equal, the amount of gas flowing through the system, and the capacity, are dependent on the absolute pressure which can be carried on the suction side. Obviously, any alteration which will permit a higher evaporator pressure will show favorable results in increased capacity, and decreased horsepower per ton. Often a close examination will show that some of the evaporator surface is partially inactive, due to insufficient flow of the air or other medium which is to be cooled over the tubes. Also, the evaporator may be so designed that a short circuiting effect takes place inside the piping or manifold. Either condition can be readily detected if sufficient temperature and flow readings are taken over the surface of the evaporator piping, and many cases are on record where a comparatively simple change in flow of the evaporating gas inside the coils, or of the medium being cooled outside the coils, produced greatly increased tonnage from the same refrigerating plant. Recently much progress has been made in the control of the liquid passing through to be evaporated, and it may be advisable to install additional or improved expansion controls, particularly on multiple systems.

If the cooled water is atomized in an air washer, a similar bypassing effect frequently reduces the available capacity of the cooling system, by returning the water to the evaporator at a temperature far below the dewpoint temperature of the air leaving the washer. In any arrangement, this spread must be as small as possible, preferably not over 5 F, and in many instances less.

It may be found that the washer pump is not handling a sufficient quantity of water, or, more often, that some of the nozzles are closed.

A form of "off peak" storage can sometimes be devised to assist the system through maximum demand periods. Space may be available for installation of a tank to hold a quantity of cooled water or other medium in reserve, or sometimes the simple expedient of establishing a method of lowering the temperature of the mass of the building and equipment by night operation will provide the effect of additional tonnage.

Controls of Major Importance

Considering the relatively short period of cooling operation, and the extremely wide variation in local factors, often from hour to hour, the controls which modulate cooling effect in accordance with the demand are of major importance. Every care should be taken to insure smooth regulation, and particular thought is warranted on the subject of properly relating inside conditions to the weather. Frequently refrigeration is found in operation without actual need for the expense involved, as when the wet bulb temperature of the outside air drops below the inside conditions. Furthermore, insufficient or improperly acting controls may reduce the inside temperature to a level 20 F below the outside—usually causing an undesirable physiological shock to persons entering the enclosure. A distinct saving may be effected by installing control devices actuated by both inside and outside conditions; in any case, recording or indicating instruments for these simultaneous readings should be provided.

The intermittent nature of the type of work provides little opportunity for all operating possibilities to be utilized, if the plan of operation for the season is not formulated with care, and therefore a comprehensive system of report forms is beneficial. The readings noted should include full data on inside and outside conditions preceding, during, and following the utilization of refrigeration effect, as well as condenser and evaporator pressures (and temperatures) entered simultaneously with dry and wet bulb temperatures on the cooling mediums.

The record will be of greater use if it contains a column for the operator's observations on the reasons for any special operation, and a parallel column for notations on the results obtained, as indicated by reports from department heads, or others in authority for the areas served. Differences of opinion can then be reconciled at their inception, rather than having them arise for discussion during the next winter, when the many advantages to be gained from them are lost.

Finally, special effort must be made to insure that the system and its operation will provide a refreshing air condition. A "canned" atmosphere is often the result where refrigeration alone is relied upon to care for the entire summer air conditioning duty. This is particularly true where odors are developed from certain processes, or where large numbers of people congregate. The preferable method is to employ the cooling effect as a controlled supplement to the basic ventilation requirement.

Windmills Protect Pipe from Corrosion

WINDMILLS are being used to protect vast pipe line networks from corrosion, according to the "Industrial Bulletin" of Arthur D. Little, Inc.

Thousands of miles of underground steel pipe carry crude oil, stabilized crude, refined products and gas. Conditions are ideal for corrosion, with heavy leakage and maintenance costs, for when pipe is thus buried underground small electrical circuits are set up, currents on which the metal floats away. Technically it is electrolytic corrosion, with the pipe the anode from which iron ions go off into the soil. One ampere of current will carry away 20 lb of steel per year, and as this loss usually comes from small local areas, the corrosion problem is serious.

By the simple procedure of reversing the current by impressing a small voltage from an outside source, this electrolytic corrosion is stopped, the pipe becomes the receiving end—the cathode—and is protected by a layer of hydrogen formed at that point.

The practice of this method may be applicable to other situations where steel is buried in moist earth. A hole is dug near the pipe line and scrap iron dumped in. Sometimes rock salt is added to make the ground more conducting. The pile of scrap iron is connected with the positive pole of a source of low voltage direct current, and the negative pole is connected with the pipe line, usually in a number of scattered places. Through the circuit thus formed, current flows from the positive pole of the source of electricity through a copper cable to the scrap iron, the anode, which corrodes rapidly but is of negligible value. The current then flows through the earth, the ions being carried by the moisture in the soil, to the pipe line, now the cathode, and a protective film of hydrogen forms on it. The current then returns to the source by another copper cable.

Any source of electric power may be used for this "cathodic protection," but the commonest are probably small low voltage generators driven by windmills.

Electrostatic Air Cleaners for Field Building

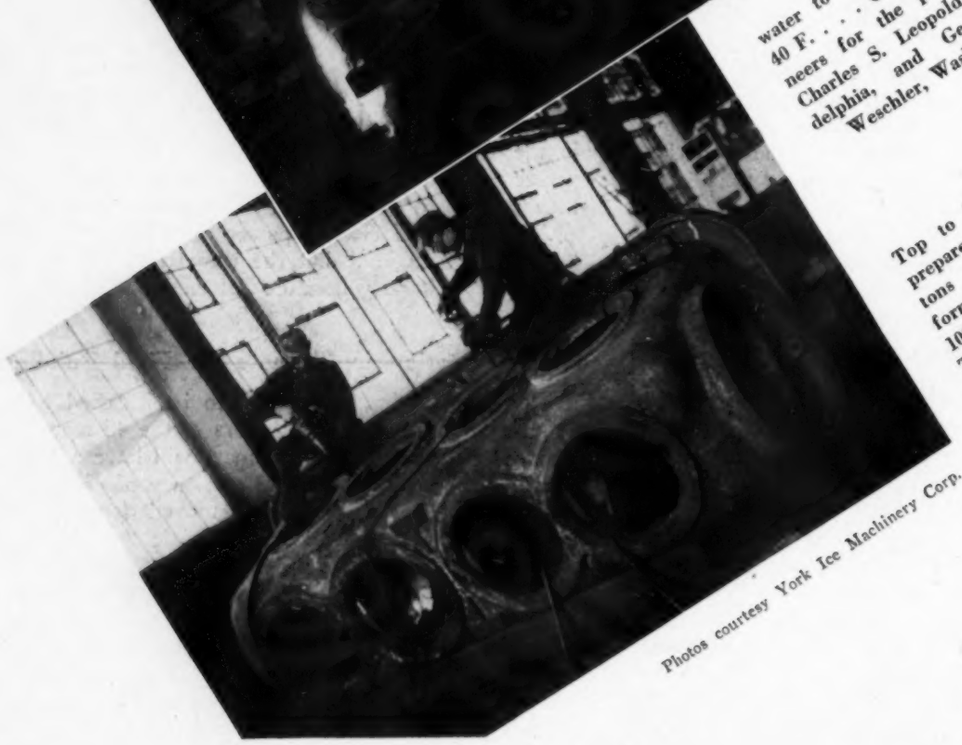
Electrostatic air cleaners are to be installed for the lower arcade and first four floors of the 45 story Field building, Chicago. The equipment will have a capacity of 272,700 cfm of air and an efficiency of over 99 per cent by weight of particles removed from the air. Operating on an electrostatic principle the air cleaner will remove microscopic particles as small as 1/5 micron in size, will operate on a current consumption of 10 kw, and will have a minimum of maintenance, it is stated.

The air cleaning equipment will consist of eighteen units ranging in capacity from 33,000 to 3,000 cfm and employing a total of 369 collector cells and 185 ionizing assemblies.

A New Idea in Air Conditioning

The Capitol power plant, now under construction in Washington, D. C., presents a new idea in air conditioning. It will house the refrigerating and water-cooling equipment for chilling water which will be piped through underground mains to air condition the Capitol, the old and new House office buildings, and the Senate office building. . . . Six four-cylinder "Freon-12" machines, each direct connected to a 1000 hp engine type synchronous motor, will provide for cooling 8000 gpm of water to a temperature of 40 F. . . . Consulting engineers for the project are Charles S. Leopold, Philadelphia, and George A. Weschler, Washington

Top to bottom—Moulders prepare a pit to receive ten tons of molten metal to form a casting for one of the 1000 hp compressors. . . . The metal is poured into the mould. . . . Workmen scale the ten ton casting



Photos courtesy York Ice Machinery Corp.

What's the Cost of Air Conditioning?

Rex E. Hieronymus* analyzes the owning and operating costs of the air conditioning system for a two story building of miscellaneous tenancy, including a bank, offices, shops, a restaurant, etc., compares the figures with the cost of a straight heating system. . . . . The method of heating and cooling employed, which is of rather unusual design, is briefly described and diagrammed

WITH the belief that there is need of as much operating data on air conditioning as it is possible to obtain, there is given here a "case history" of the costs for one specific example—a completely air conditioned building which has been run for more than a year with normal occupancy. The building is the two story, taxpayer type structure with basement erected at Monroe and Dearborn streets, Chicago, during the winter of 1934-1935.

Designed by Holabird & Root, architects, it is an L shaped building fronting on two intersecting streets. Except for the entrance architrave, the two street elevations are entirely of glass and steel, the latter in the form of two continuous bands of stainless steel, one at the top of the first story and the other from the top of the second story windows to the top of the parapet. The second story is set back 4 ft-6 in. from the building line.

The owners decided to install a complete air conditioning system because they believed it would stimulate renting and return its cost by higher rentals in a period of twelve years, the contemplated life of the structure. The first floor was to be developed for miscellaneous commercial tenants and when the building was designed the north wing of the second floor was leased to a bank. This lease probably was contingent upon air conditioning, and representing as it did approximately 25 per cent of the space, it was an important factor in the decision to air condition. The south wing of the second floor was to be offered for miscellaneous small commercial offices. Steam for heating and other purposes is purchased from the local utility.

Partial List of Mechanical Equipment

Refrigerating Plant:
Centrifugal water vapor compressor, capacity 105 tons.
Driven at 9230 rpm through gear unit by induction motor, 100 hp, 1755 rpm, volts 208, amps. 244, 3 phase.
Accessories: 715 sq ft surface condenser; evaporator tank integral with base of unit; chilled water pump and vacuum pump. Pumps driven by 15 hp and 10 hp motors respectively.
Heat Exchanger:
3½ million Btu per hr.
Pumping Plant:
Circulating pumps, two, 225 gpm. Each driven by 3 hp motor.
Air washer pumps, three—two 170 gpm, one 52 gpm. Driven by 5 or 2 hp motors.
Air Washer Units: Each consisting of:
Preheater and reheater coils.
Filters, throw-away hair type.
Spray nozzles, flushing type.
Eliminators.
Ventilating Fans:
Six supply fans, totaling 51,400 cfm, of which 45,400 cfm is conditioned air.
Seven exhaust fans, totaling 47,700 cfm.
Temperature Control:
Low voltage a-c.
Store Coolers:
Eighteen, suspended type.
Office Coolers:
Thirteen, floor type.

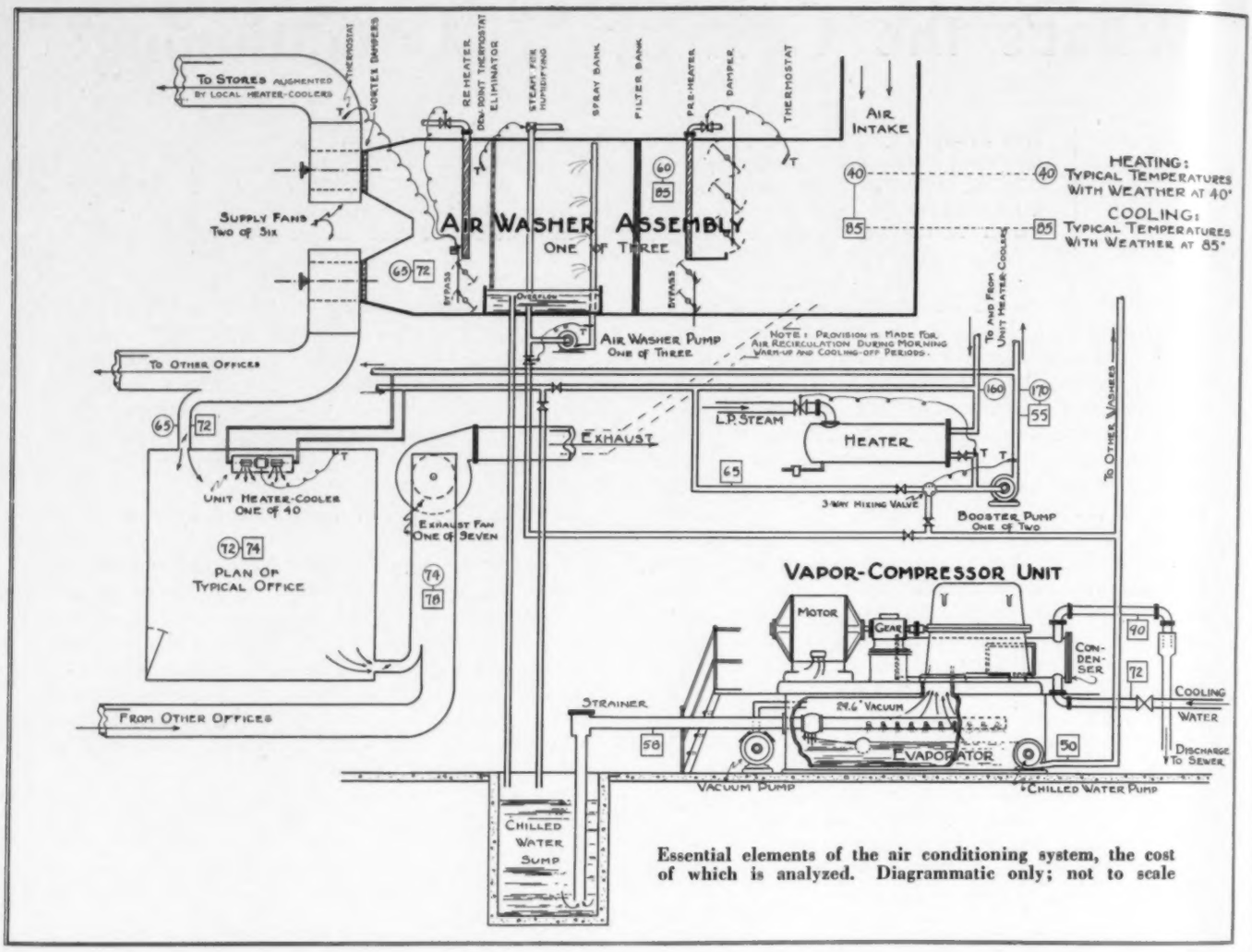
*General Superintendent, Aldis & Co., Real Estate.

The Type of Air Conditioning System

The choice of type of summer cooling system and the selection of equipment was given thorough study, and it was decided to install a combination system of new design so constructed that the major apparatus could be used both summer and winter. It was decided to ventilate the entire building the year 'round by mechanical means and this permitted the use of fixed glass in the window surfaces and openings. The system is designed to heat the building to 72 F with full code ventilation at an outside temperature of —10 F, and to maintain an indoor relative humidity from 50 per cent in milder winter weather down to the highest humidity that can be maintained without frosting the windows in more severe weather. For summer operation the system has capacity to produce a 13 deg differential between outdoor and indoor temperature when the outside conditions are 95 F dry bulb and 39 per cent relative humidity, resulting in 50 per cent inside relative humidity.

The year 'round ventilation is accomplished through three air conditioning units in the basement with a common air intake carried down from the second story rear. Each of the units comprises (in the order named) steam tempering coils, filters, air washing section with spray nozzles, reheating steam coils, and supply fans. For each of the main supply fans there is a corresponding exhaust fan. Where permitted by the code, recirculating connections are provided between the supply and exhaust systems. This arrangement constitutes a type of zoning and the different sections can be operated at different times according to occupancy requirements.

Heating in winter is accomplished by steam coils, and hot water heated in a steam heat exchanger; in summer, cooling is effected by water chilled by a centrifugal water vapor compressor rated at 105 tons and driven by a 100 hp induction motor. Because the exact character of the tenants for the greater part of the building was not known at the time the system was designed, it was decided to incorporate maximum flexibility to handle future requirements by arranging the system to provide for the summer cooling and winter heating to be accomplished in two stages. Under this plan the main



Essential elements of the air conditioning system, the cost of which is analyzed. Diagrammatic only; not to scale

or first stage of cooling or heating is done in a central system of apparatus situated in the basement and the supplemental or second stage is done by local unit heater-coolers situated directly in the various conditioned spaces.

It might be said that the first stage handles the treatment of the air necessary for ventilation and the summer constructional heat gain load and that the second stage handles the internal heat gain in summer and a portion of the constructional heat loss in winter. Thus about half the heating effect in winter and half the cooling effect in summer is accomplished by circulated air conditioned in the air washer assemblies. The other half of the heating or cooling load is handled by the unit heater-coolers through which chilled water is circulated in summer and hot water in winter. (The heating of the bank is augmented by cast iron radiators installed below the windows, and supplied with circulated hot water the same as the unit heater circuits.) The proper division of the water among the various circuits is obtained by balancing valves in the branch lines leaving the main header.

The unit heater-coolers in the stores are for the most part suspended from the ceiling at appropriate locations; on the second floor they are of the floor type, in front of the windows. Each unit is controlled by a reversible thermostat in the space it serves, giving thermostatic control both summer and winter and making up, from each unit, the deficiency of heating or cooling not provided by the circulated air. Thus, there is provided year 'round air conditioning, consisting of heating, humidifying, filtering, washing, and circulation in winter, and cooling, dehumidification, filtering, washing, and circulation in summer.

The building was formally opened on March 18, 1935, with partial occupancy, the bank, first floor restaurant, and one or two other first floor stores being occupied. As rapidly as leases could be negotiated and alterations completed, more tenants were installed and by September 15, 1935, the building was entirely filled except for approximately 1200 sq ft of second floor space. Since the rentable area of the building is 39,187 sq ft, there has been an occupancy ratio of 96.5 per cent during the past year or more.

Putting the Plant in Operation

Beyond usual difficulties incidental to placing a rather complicated mechanical plant in operation, there has been no real trouble either summer or winter with the completed system. In the tavern on the first floor difficulty was had in preventing uncomfortable drafts be-

cause of the high heat load from occupants, steam tables, and electric lighting above the average in power consumption. Rearrangement of the grilles corrected the trouble. Changes had to be made in location of duct thermostats and in the arrangement of certain dampers but these difficulties were overcome so that satisfactory operation of these features is had in all weather, varying from the most severe through the difficult marginal weather.

One difficulty which it seems there is no possibility of avoiding occurs in marginal weather when occasionally some tenant will want heating and some other tenant cooling from the same air conditioning unit at the same time. On these occasions the engineer compromises the conflicting requests and does as he thinks best.

There is no question but that complete air conditioning of a building is of value in renting space, although there is some unwillingness on the part of some occupants to pay in rent what the service actually costs. There is a further disposition on the part of some of the stores which handle standard merchandise that the purchaser can select easily and quickly, to leave the store door open in hot weather, claiming that it is more favorable to trade. Such a practice denies them the benefits of proper cooling and upsets the operation of the central system, causing some difficulty unless they can be persuaded not to do it.

Cost Analysis of Air Conditioning

While it is easy to divide the operating commodity costs between the heating and cooling seasons, it is difficult to devise a means of dividing rationally the total costs, for the reason that the capital investment is rather high, and although it was necessitated mainly on account of the summer cooling program, nevertheless most of the plant is also in use during the heating season. It would seem perfectly proper, then, to divide the owning costs between the two seasons. However, we must remember that the building if not air conditioned would merely have been heated in winter with some standard system, probably cast iron radiation, and except perhaps for dead-end stores or special premises such as restaurants, it would not be mechanically ventilated. Therefore it appears fair to assign the owning cost of the special plant to summer air conditioning.

The complicated plant has also imposed higher operating costs during the winter season both for extra steam required to heat the larger volume of ventilating air and for labor. Electric power to drive the fan systems and circulating pumps is also an item and supplies and repairs are higher than for a simple direct radiation system.

Really to compare the cost of completely air conditioning the building with the costs customarily experienced in a conventional building, it seems proper to determine on an annual basis the present costs (labor, steam, electricity, and water) for heating, ventilating, and cooling the building, and to compare them with the corresponding costs in a building having conventional winter heating and no summer cooling. The cost for the latter type of building can be rather accurately estimated from experience in similar buildings and this has been done. The tabulations shown are offered as an accurate determination of the cost of complete year 'round air conditioning of this particular building.

The total of the costs tabulated indicates an annual cost of commodities (steam, water, and electricity), supplies and repairs, totaling $8876, or 22.6c per sq ft of rentable area. To this must be added the cost of labor to operate the system summer and winter. It appears from a rather detailed study that complete air conditioning entails an annual labor cost to the building of $5910, or 15.1c per sq ft of rentable area. The total operating cost of the complete air conditioning, before fixed charges on the plant, therefore totals $14,786 per year, or 37.7c per sq ft.

Wishing to know the full capital cost of air conditioning the building, to the cost of the heating and ventilating contract was added the cost of the plumbing, electrical, and general contractor's work which was made necessary by the conditioning system. This was $93,258 including fees. The cost of a straight direct radiation heating system with necessary ventilating equipment for the first floor and part of the basement was estimated to be $28,376 with fees. The difference, $64,882, may be called the extra capital cost of complete air condi-

Heating Season—October 1, 1935, to May 15, 1936

1. Average temperature during period, F	37.6
2. Number of days with average outside temperature of zero or below	10
3. Number of days when minimum temperature was zero or below	25
4. Number of degree-days (normal is approx. 6000)	6770
5. Heating steam cost, 4,559,000 lb at 0.985 per M lb	$4489
6. Electric power cost for pumps and fans, 79,579 kwhr at $0.01715 per kwhr	$1365
7. Supplies	$ 134
8. Repairs	$ 15
9. Labor (not included)	....
Total, Items 5-8 inclusive	$6003
Cost per degree-day	$0.887
Cost per sq ft of rentable area	$0.154

Cooling Season, May 16, 1936, to September 30, 1936

1. Average temperature, F	69.8
2. Number of hours main compressor was operated	1227
3. Average outdoor temperature during these hours, F	78.5
4. Average indoor temperature during these hours, F	74.0
5. Number of days cooling was used	99
6. Number of days above 80 F average outdoor temperature	22
Number of days when maximum outdoor temperature was 80 F or above	59
7. Electric power cost for compressor, 89,284 kwhr at $0.0157 per kwhr	$1402
8. Water cost for condensing purposes, 1,077,000 cu ft at $0.51 per M cu ft	$ 549
9. Electric power cost for fans and pumps, 51,540 kwhr at $0.0157 per kwhr	$ 809
10. Supplies	$ 113
11. Repairs	0
12. Labor (not included)	....
Total, Items of Expense	$2873
Cost per sq ft of rentable area	$0.0733

tioning over straight heating and partial ventilating. Dividing this figure by 39,187 sq ft gives a cost of $1.65 per sq ft of rentable area, which may be called the extra installation cost due to year 'round air conditioning.

For brevity's sake the various tabulations are omitted, but it is found that the owning cost of the complete plant as installed is $13,378 annually, or 34.2c per sq ft per year. Adding to this the operating costs, including labor, gives a yearly cost for complete air conditioning of $28,164, or 71.9c per sq ft of rentable area.

It is found that the corresponding cost for this building without air conditioning, but with winter heating, would be $9,390 annually, or 24c per sq ft of rentable area per year.

The difference, $18,774, represents (on a 12 year basis) the extra annual cost of furnishing complete air conditioning service, and it amounts to 48c per sq ft of rentable area per year. If this figure seems high, it must be remembered that this amount covers the charges for year 'round air conditioning and is not to be confused with costs for "summer cooling" only, which is also popularly referred to as "air conditioning". It should also be realized that the building analyzed probably represents an unfavorable case from a cost standpoint, partly for the following reasons:

There is considerable diversity in the character of different tenancies and these diversified businesses have incurred larger first costs for installation of the air conditioning system over what would be required in a single-purpose building. The higher installation cost is reflected in the owning costs.

The nature and size of the building happen to be such that the attendant labor expense chargeable to air conditioning is higher than would be the case in a larger building, for it appears probable that the present help could take care of a plant, say, three times as big, which of course would reduce proportionately the charge per sq ft for operating labor.

It may be thought that the proposition should have been figured on a period longer than 12 years. That term was taken because this whole taxpayer building was predicated on a 12 year life. However, it is not uncommon to figure air conditioning systems for a life of 15 years or more, and were this done in the present case there would be a further proportional reduction in the owning cost.

Gas as a Source of Power for Air Conditioning

By B. A. Johnson*

WHILE the idea of using a gas engine as a source of power for refrigeration for air conditioning is not new, it is only with the comparatively recent development of engines designed to burn gas that this type of power has reached a competitive level with purchased electric power and diesel power.

Before the advent of what we might call the "gas designed" engine, units designed to burn gasoline fuel (both automotive and industrial type) were converted to burn gas. This is accomplished by changing the method of carburetion and also, in the case of automotive engines, reducing the working speed of the engine. Naturally, such conversion results in loss of efficiency and is only feasible where there is a source of very cheap gas for fuel. Good operation for this type of converted unit is about 13,000 Btu per bhp-hr using 1000 Btu natural gas.

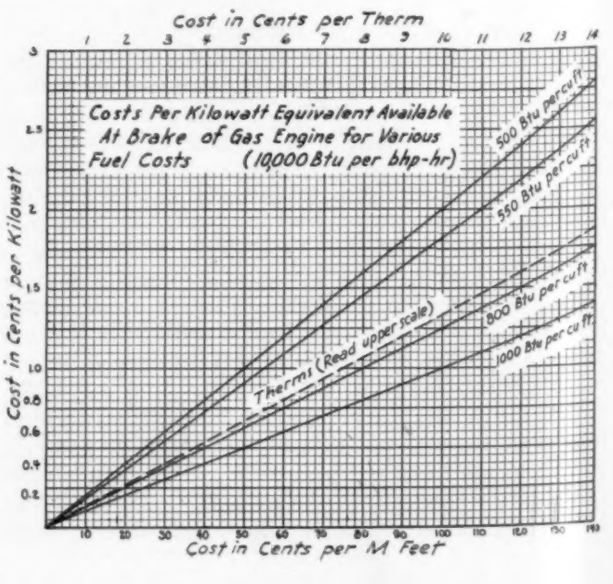
As the possibilities of a "gas designed" engine began to be realized, active development work along the lines of higher compression ratios, separate thoroughly water cooled valve-in-head cylinders, independent air and gas mixing valve controls, etc., produced gas burning engines capable of developing at least 20 per cent more power than gasoline engines of the same size. Good operation of a gas designed engine today gives a fuel economy of about 10,000 Btu per bhp-hr.

Today gas engines are available in sizes ranging from 10 to 400 hp or more. For approximate cost estimating, the figure of $40.00 per bhp will give an average first cost. Weighing on the average 75 lb per bhp as compared, for instance, to 28 lb in a gasoline engine, gas engines are of extremely rugged construction and will last at least 20 years with intermittent operation.

A simple way by which to compare the cost of gas engine power with electric power is to express the cost of delivered gas engine power in terms of kw for any given cost of gas. By comparing such a figure with the electric rate per kwhr we get a very close comparison of the relative cost of power available for a given cooling or refrigerating job (see chart.)

For example: Gas at 7c per therm will produce available energy for cooling at 0.94c per equivalent kw, 550 Btu gas at $1.00 per M, 1.82c per equivalent kw. Comparing these equivalents with any prevailing rate for purchased electrical power then gives a comparison of fuel costs for performing a given refrigerating job.

It should be pointed out that the costs shown by this chart refer to gas engine power which can be connected directly to any cooling or refrigerating machine as compared to electric power which must be connected to mechanical energy through an electric motor.


Cost in Cents per Therm
Costs Per Kilowatt Equivalent Available At Brake of Gas Engine for Various Fuel Costs (10,000 Btu per bhp-hr)
500 Btu per cu ft
550 Btu per cu ft
800 Btu per cu ft
1000 Btu per cu ft
Therms (Read upper scale)
Cost in Cents per Kilowatt
Cost in Cents per M Feet

*The Peoples Gas Light and Coke Co. From a paper presented at the University of Wisconsin conference on air conditioning and automatic heating, April, 1937.

E. Vernon Hill* Comments on....

.....Air Conditioning and the Dionne Quintuplets

TWENTY-FIVE years ago the consulting engineer, pioneering in what we now call comfort cooling, had an easy time of it compared with the engineer of today. True, he had to be ingenious, resourceful, and well-grounded in the fundamentals of thermodynamics. But there was only one refrigerant to be seriously considered, CO_2; one refrigerating machine and one type of apparatus, the baudelot cooler, for transferring the refrigerating effect to the air stream; one method of air cleaning, the water spray; one system of air distribution, the trunk line system with a few individual modifications; and one system of automatic control, the pneumatic.

Look at him today. "Freon" has crowded into the spotlight. At least three new refrigerants have been developed and accepted for centrifugal compressors operating below atmospheric pressure.

Next the vacuum or steam jet machine using water as a refrigerant was refined, and its efficiency greatly improved. Today the absorption method with a new refrigerant, dichloromonofluoromethane, is knocking at the door.

And then the fin type coil for both heating and cooling, and direct expansion applications in a multitude of different forms and designs, has been developed. Air filters, a score of designs, cell type and automatic, crowd the engineer for recognition and acceptance. The old trunk line system of air distribution must be modified and redesigned to be accommodated to the high velocity jet discharge and the new resistance and diffusion grilles. Suspended unit coolers, spot coolers, and new models of unit distribution appear daily.

Temperature control is no longer the simple process of producing motion by relieving the pressure in a valve or damper motor with a bimetallic thermostat. Electric controls have invaded the field and must now also be considered by the engineer in practically all forms of heating and cooling apparatus.

Specialization must be resorted to or the independent designing engineer will be crowded from the picture. What a wonderful opportunity for the Dionne quintuplets in this important engineering field had they only been born boys.

.....Air Conditioning and Moses

THE dissolution of old ventilation practice and the evolution of air conditioning has brought a situation which has caused the industry many a severe headache. Everything is now labeled "air conditioning," from a furnace fan to the most complete heating, air cleaning, and refrigerating equipment. The ambitious labels, however, are not particularly serious; it is the wide variation in performance guaranteed or obtained by different kinds of equipment, and the inexcusable spread in the prices quoted, that causes criticism and discord.

In an endeavor to correct this situation several societies and interested groups are endeavoring to frame air conditioning codes or standards that will bring at least a measure of order out of the chaos, and place designs and bids on specifications on a somewhat more understandable and comparable basis.

It is, however, no easy task to draft a code telling in sufficient detail how to figure the refrigeration load, what factors to use, how much air shall be circulated, what size compressor to install, how to design the duct system, what volume of air must be handled, etc., etc., *ad infinitum*.

It is difficult because air conditioning is a new and vital industry; it is still in the formative stage and the good practice sponsored by a committee today is obsolete in many details tomorrow. Furthermore, such air conditioning codes, to be effective, must be subscribed to and enforced by all branches of the industry, the architect and engineering organizations, the steamfitters, sheet metal contractors, and the manufacturers of equipment.

Their efforts are sincere but their viewpoints are necessarily at variance, and so the situation becomes complicated. It will not be surprising if complete acceptances of such codes are delayed by conflicting ideas and interests, until air conditioning practice *of itself becomes standardized*, and then no code will be required.

Several thousand years ago Moses came down from the top of Mount Sinai with two tablets of stone on which were engraved ten commandments. They were intended for the guidance of the Jewish race recently emancipated; not in some minor activity like air conditioning but in their entire individual and communal relations. The commandments were couched in language that was direct but general. They did not undertake to tell the Israelites what to do but what *not to do*. "Thou shalt not" was the phraseology used, and there were only ten.

I suggest to the various code committees studying air conditioning problems to take a tip from Moses—write briefly and to the point a few things in general that should be avoided, that *should not* be done. It is possible that if this procedure were used, codes could be prepared and adopted more promptly, and would consequently be more valuable.

*Member of Board of Consulting and Contributing Editors.

Proper Installation *of* Centrifugal Pumps

INSTRUCTIONS for operating centrifugal pumps from the new standards issued by the *Hydraulic Institute* appeared in these pages last month. The following information on installing such pumps has been taken from the same source, with the permission of the *Institute*.

Location of Unit

The pump should be as near the water supply as practical so that the static suction lift will be low and so that a short direct suction pipe may be used. The pump should be located so that a short direct discharge pipe with the least number of elbows and fittings may be used, to minimize head loss from friction. It should be placed so that it will be accessible for inspection during operation. Head room should be provided when it is necessary to use cranes. The pump should be protected against floods and a motor driven unit should be installed in a dry place if possible.

The foundation should be sufficiently substantial to absorb any vibration and to form a permanent, rigid support for the base plate. This is important in maintaining the alignment of a direct connected unit. A concrete foundation on a solid base should be satisfactory. Foundation bolts of the proper size should be embedded in the concrete, located by a drawing or template.

It is recommended that the services of a manufacturer's erecting engineer be employed in installing and starting up pumping equipment which is costly or of a precision type. This insures the buyer that the machinery is properly installed and also affords his men the privilege of receiving special instructions.

Alignment

It is recommended that manufacturers do not dowel pumps and drivers on bedplates before shipment. The pump with driver is correctly aligned on bedplate before shipment. This alignment, however, may be disturbed during transit or by incorrect leveling of bedplate on foundation. If the dowel pins are inserted before shipment, undesirable springing of the bedplate may be necessary to re-establish original alignment. It is consequently advisable to dowel pump and driver only after alignment has been established on foundation.

With a pair of inside calipers or a thickness gage, the distance between the coupling halves should be checked at four points. (See Fig. 6.)

If the pump is to be connected to the prime mover by gears or chain, check the alignment by the use of a straight edge across the faces of the gears or sprockets. This should be done in two directions at an angle to each other as large as permitted by the relative size of the gears or sprockets.

Whenever the pump or prime mover, or both, are heated in operation (that is, steam driven prime movers, or pumps handling hot water), the unit should be aligned under these conditions. This is necessary due to changes in alignment on account of expansion from the cold to the hot condition.

Any base plate, no matter how heavily it is built, may be slightly sprung in shipment, or may be distorted by an uneven support on the foundation, or by uneven tightening of the foundation bolts, or by the pull from the pipe connections. It is necessary to be careful when installing the pump to secure perfect alignment of the coupling. A flexible coupling will not compensate for misalignment. Inaccurate alignment of the coupling results in rapid wear of the coupling bushings, heating of the bearings and loss of efficiency. Therefore, after the pump is fastened on the foundation it is necessary to see that the shaft of the pump and of the prime mover are in one line. If the prime mover and pump are direct connected, remove the coupling bolts, if not already removed. The pump should be completely connected up to its piping and the base plate leveled up then, and adjusted to position by shims and wedges so as to bring the two halves of the coupling into perfect alignment. (See Figs. 5 and 6.)

Grouting

For grouting, build a dam around baseplate at least 2½ in. high.

After cement has hardened, tighten the anchor bolts and recheck alignment. Any misalignment now apparent should be corrected by placing shims under pump, motor or brackets.

Piping and Valves

Both suction and discharge pipes should be independently supported near the pump so that when the flange bolts are tightened no strain will be transmitted to the pump casing.

It is usually advisable to increase the size of both suction and discharge pipes at the pump nozzles in order to decrease the loss of head from friction, and for the same reason piping should be arranged with as few bends as possible and these should be made with a long radius wherever possible.

Discharge Piping—A check valve and a gate valve should be installed in the discharge line. The check valve, placed between the pump and the gate valve, is to protect the pump from excessive pressure and to prevent water running back through the pump in case of failure of the driving machine. The gate valve is used in priming and starting and when the pump is to be shut down. It is advisable to close the gate valve before stopping the pump when a foot valve or check valve is installed to prevent water hammer. This is especially important when the pump is operated against a high static head. If reducers are used on the discharge side to increase the size of discharge piping, they should be placed between check valve and pump.

Suction Piping—The suction pipe must be kept free from air leaks. This is particularly important when the suction line is a long one or the static suction lift is high. (See Figs. 1-4.)

Trouble is often caused by the use of bell and spigot pipe in the suction line. Screwed or flanged pipe for the smaller sizes and flanged pipe for the larger sizes or for high suction lifts are recommended.

The suction pipe should slope upward to the pump nozzle. A horizontal suction line must have a gradual rise to the pump. If other piping is in the way, go under it. Any high point in the pipe will become filled with air and thus prevent proper operation of the pump. A straight taper reducer should not be used in a horizontal suction line, as it forms an air pocket in the top of the reducer and the pipe. Use an eccentric reducer instead. If an air pocket is left in the suction pipe when the pump is primed, the pump will often start and operate properly for a time, but it is probable that air will be drawn from the air pocket and up into the pump, causing the pump to lose its prime.

This is especially true when the pump is primed by the use of a foot valve. But even when it is primed by ejector or vacuum pump a small quantity of air is left in the pocket and this is added to, both from air in the water released by the partial vacuum in the suction pipe and from air admitted through minute leaks in the pipe connections that ordinarily would not cause trouble.

Small air pockets that may cause trouble are formed in the top of any gate valves installed vertically in the suction line. It is recommended that the gate valves in the suction line be laid so that the stems are horizontal. Trouble caused by an air pocket in the suction line can usually be stopped temporarily by priming and starting the pump several times. This will draw out enough of the air in the pocket so that the pump will operate properly, but the trouble is liable to recur, and such a pocket should be eliminated.

Check Valves—Check valves should ordinarily have no place in the suction line, although they are sometimes used in series-parallel operation to reduce the number of valves which are to be operated when changing from series to parallel operation. The pump must not be throttled by the use of a valve on the suction side of the pump.

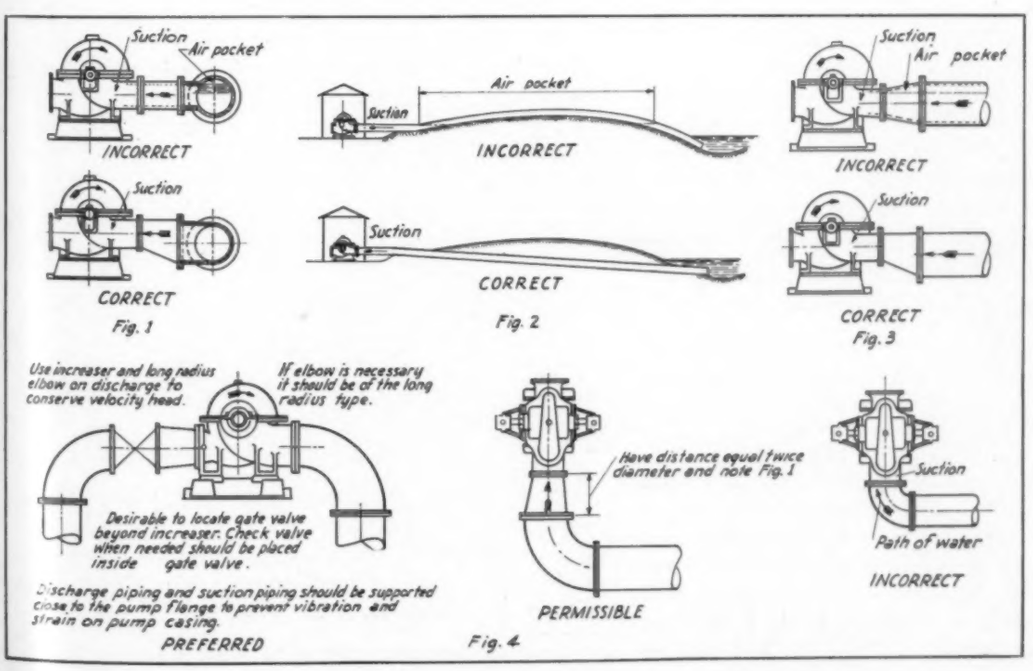
Elbow on Suction—Suction piping for double suction pumps should not be installed so that there is an elbow close to the suction nozzle of the pump except when this elbow is in a vertical position; that is, an elbow bending either straight up or straight down is permissible, but one entering at another angle should not be used. There is always uneven flow around an elbow and when it is installed in any but a vertical position this unequal flow causes more water to enter one side of the runner than can enter the other side, thus causing a reduction in capacity and efficiency and a thrust which will heat the thrust bearing and possibly be of sufficient magnitude to cause rapid wear of the thrust bearing.

Foot Valve—When the suction lift is not very high it is frequently advisable to install a foot valve to facilitate priming. Foot valves should not be used when the pump is operating against a high static head, as failure of the power would allow the water to rush back suddenly, causing a heavy water hammer. The foot valve should be of the flap type rather than of the multiple spring type, with a clear passage for water of at least the same area as that of the suction pipe.

Care must be taken to prevent foreign substances from being drawn into the pump or choking the foot valve, and for this purpose an efficient strainer should be provided. When there is any refuse, such as sticks, twigs, leaves, etc., in the water a larger outside screen should be placed around the suction inlet to prevent choking the strainer. This screen should have sufficient openings to keep the flow through it below two feet per second.

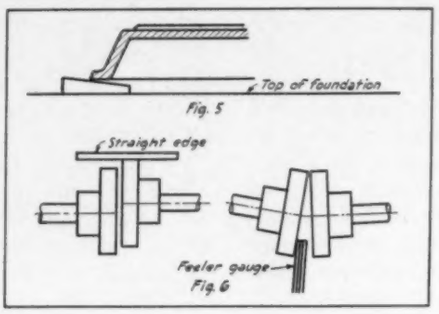
Stuffing Boxes and Packing

The stuffing boxes are ordinarily packed before shipment. If, however, the stuffing boxes are not packed, they should be carefully cleaned and the packing placed in them. Be sure that sufficient packing is placed back of the water seal ring so that the water for sealing is brought in at the water seal ring and not at the packing. The piping supplying the sealing water should be fitted tightly so that no air is brought in, as a small quantity of air entering the suction at this point may result in the



pump losing its suction. If the water to be pumped is acid, dirty or gritty, sealing water should be piped to the stuffing boxes from a clean outside source of supply in order to prevent damage to the packing and shaft sleeves.

In placing the packing, each packing ring should be cut to the proper length so that the ends come together but do not overlap. The succeeding rings of packing should be placed in the stuffing box so that the joints of the several rings of packing are staggered. The packing should not be pressed up too tight, as this may result in burning the packing and cutting the shaft. The stuffing box is not properly packed if friction in stuffing boxes prevents turning the rotor by hand. On starting the pump it is well to have the packing slightly loose without causing an air leak and, if it seems to leak, rather than put too much pressure on the gland, put some heavy oil in the stuffing box until the pump works properly and then gradually tighten up the gland. The packing should be occasionally changed.



Bearings

Oil Lubricated Sleeve Bearings—Before starting the pump clean the bearings thoroughly, as dirt and foreign substances may get in during shipment or erection. The bearings should then be filled with a neutral mineral oil of about 300 seconds Saybolt viscosity at 70 F. This oil should be changed when it becomes dirty and the bearings cleaned out at the same time. Bearings should be examined for wear at frequent intervals. When the pump is first started, the operator should make sure that the oil rings are turning freely. They may be inspected through the oil holes in the bearing caps. During the first hour of operation the bearings should be watched for overheating.

Ball Bearings—It is impossible to over-emphasize the importance of proper lubrication of ball bearings in pumps.

Oil lubricated ball bearings may be lubricated with the same grade of oil as used for ring oiled sleeve bearings. It is recommended for grease lubrication that regular ball bearing grease be used. Nearly all oil companies have a standard grease for ball bearings, or standard commercial Vaseline can be substituted. Do not use graphite. Heating of bearings invariably is caused by too much instead of too little grease, and careful inspection to determine the trouble should be made before more grease is added.

Great care should be exercised to keep housing clean and only clean grease should be used. Foreign solids or liquids invading the bearing housing can completely ruin the bearings in a short time. When cleaning ball bearings, flush with gasoline or kerosene and use clean instruments and cloths. It is impossible to state how often ball bearings should be greased or cleaned; this must be determined by the installation and the service.

Roller Bearings—Roller bearings should be lubricated with a short fiber soda-base grease or with a high grade Vaseline or petroleum. Greases containing graphite or a filler should not be used. Greases graded number three or four are usually too heavy. It is impossible to say how often grease should be added, as this depends on the condition of operation. It is well to add two or three ounces of grease at a time at regular intervals until the operator can tell how often more grease will be needed. Never pack the bearing housing full, as the excess will run out on the shaft.

Miscellaneous

It sometimes happens that there is a considerable amount of air or gas in the water pumped. If there is much of this, there may be a tendency for the gases to separate in the passages of the pump. For this reason air cocks placed on top of the casing should be opened occasionally to allow the air to escape. If there seems to be an unusual amount of air, these vents may be opened slightly at all times during operation of the pump, connecting them to drain pipes if desired.

Care should be taken to prevent the pump from freezing during cold weather when it is not in operation. It may be necessary, when there is any possibility of this, to drain the pump casing of its water during the shutdown period by removing the plugs in the bottom of the casing.

Improving Heating Plants

In the improvement of heating plants there seems to be no question but that too much caution has been exercised. However, as a general precaution, it has seemed wise, at least in this operator's experience, to go after these improvements on a long-time basis and to make only one major change in a department at a time, pursuing each change until definite improvement has been obtained from each move. However, if one improvement be in the steam production department and the other improvements be in the upstairs utilization, both these changes can safely be instituted at the same time. Keep your program so arranged, however, that you can actually determine results from any particular move and get the most benefit; otherwise you run the risk of doing three or four things at once and finding yourself pleased at, say, a 10 per cent reduction in costs, whereas the net improvement really should be 40 per cent. In such a case it is well to go after the improvement 10 per cent at a time, so to speak.

Your program for improvement should never be completed. When you have accomplished the last item on your original list of possible improvements, go back and review your early improvements again with a dispassionate eye.—Rex E. Hieronymus.*

*General Superintendent, Aldis & Co., Real Estate.

How to Avoid Trouble on Cooling Jobs With Evaporators Above Condensers

[Part 2—Selecting Expansion Valves]

By William Goodman*

Where it is necessary to place the evaporating coils some distance above the condenser or receiver—as it is in many air conditioning installations—problems are created which must be recognized and properly dealt with if trouble is to be avoided. Among the difficulties are those due to incorrect selection of the expansion valve. . . . Here is a thorough discussion of the problem involved and an explanation of what to do about it

WHERE it is necessary in an air conditioning or refrigerating installation to place the evaporating coils some distance above the condenser or receiver, problems are created which must be recognized and properly dealt with if subsequent difficulties are to be avoided.

Among the principal difficulties are those due to the incorrect selection of expansion valves. Valves must be chosen not only for the smaller pressure difference, due to the lower pressure existing at the top of vertical pipe lines, but their selection must also allow for the small amount of vapor formed in the vertical line. Although the actual weight of vapor is small, the volume occupied by it is relatively large—so much so that it has a substantial effect upon the capacity of an expansion valve.

Part 1, published last month, discussed the pressure variation in a vertical "Freon" pipe line, how to compute the pressure at any point, and how to insulate the piping. Selection of expansion valves is our subject this month.

Rating of an Expansion Valve

An expansion valve (like any other valve) consists of nothing more or less than an orifice with a device for adjusting the flow of fluid through that orifice. The weight of liquid that can flow through the orifice when the valve is wide open depends entirely upon the difference in pressures before and after the valve. However, the capacity of an expansion valve is not rated in terms of the weight of liquid passing through it but in terms of the refrigerating capacity of the liquid. Thus a valve is said to have a capacity of 10 tons at some given pressure only because it can pass such a weight of liquid which, vaporizing in the coils, will absorb the heat equivalent of 10 tons of refrigeration. In selecting expansion valves, therefore, it is always necessary to know the pressure difference across the valve. Valves cannot be selected on the basis of the refrigerating capacity alone. Expansion valves should be selected for the actual pressure difference at which they will be required to pass the necessary weight of refrigerant.

In a perfect system, the pressure at the discharge of an expansion valve would be equal to the pressure being maintained in the evaporating coils; in actual systems, there is always some pressure loss in the distributing headers on the coils. For this reason the pressure at the discharge of an expansion valve will always be 5 to 10 lb higher than the pressure in the evaporator.

The pressure at the inlet of an expansion valve can be computed by means of Fig. 2, published in Part 1 last month. The final pressure obtained from Fig. 2 is based only on the loss of pressure due to the vertical rise of the liquid line. In any actual system there will also be a pressure loss due to the friction of the mixture flowing in the pipe line. The computation of the pressure drop of liquid "Freon" in pipe lines would require an article in itself, but in the usual system will amount to a total of about 2 or 3 lb, depending of course upon the size of the pipe.

Example 3: Take the conditions of Example 1 and compute the difference in pressure across the expansion valve if the pressure in the evaporator is 37 lb ga (corresponding to an evaporator temperature of 40 deg). The evaporator is situated 40 ft above the liquid receiver. Assume that the pressure drop through the liquid header on the evaporating coils is 7 lb and that the friction loss in the liquid line is 2 lb.

Solution: Pressure at lower boundary (From Example 1) ..... 116.9 lb ga

Vertical distance from receiver to evaporator..... 40.0 ft
Distance from receiver to lower boundary (From Example 1)..... 13.3 ft

Distance between lower boundary and evaporator ..... 26.7 ft

Referring to Fig. 2, for a lower boundary pressure of 116.9 lb and a vertical distance of 26.7 ft,
Final pressure at top of riser = 105 lb ga.
Since there is an additional pressure loss in the liquid line of 2 lb due to friction,
Pressure ahead of expansion valve..... 105 — 2.0 = 103 lb ga

Pressure in the evaporator..... 37 lb ga
Pressure drop through liquid header..... 7 lb ga

Pressure required at discharge of expansion valve ..... 44 lb ga

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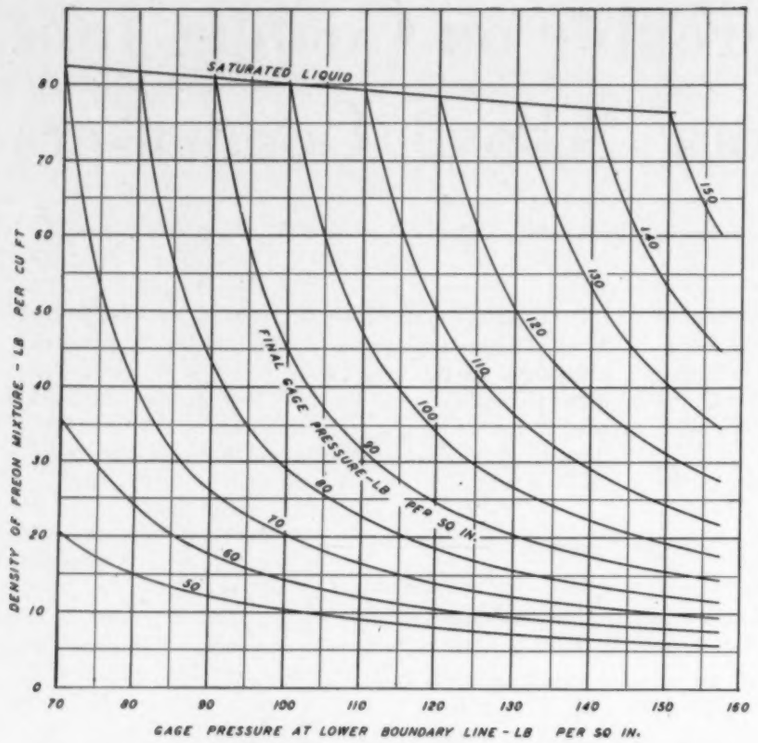


Fig. 3—Chart for finding the density of mixtures of liquid "Freon" and vapor. The gage pressure at the lower boundary line (see Part 1) and the final gage pressure must both be known

Pressure difference across expansion valve = 103 — 44 = 59 lb per sq in.

For the conditions of this example, the expansion valve should be selected for a pressure difference of 59 lb per sq in.

Effect of Refrigerant Density on Valve Capacity

Expansion valve capacity is lowered not only by a reduction in the pressure difference but also by a reduction in the density of the refrigerant. Since a portion of the "Freon" vaporizes in the liquid line, there is evidently a marked reduction in the density. Fig. 3 gives the density of mixtures of "Freon" liquid and vapor after having been expanded from any given lower-boundary pressure, to some final pressure ahead of the expansion valve. (The method of computing the curves of Fig. 3 will be given in the appendix.)

Example 4: Take the conditions of Example 3 and find the density of the liquid ahead of the expansion valve.

Solution: From Example 3,

Pressure at lower boundary.....116.9 lb ga
Final pressure ahead of expansion valve.....103.0 lb ga

Referring to Fig. 3, for the above conditions,

Density of mixture of "Freon" liquid and vapor = 43 lb per cu ft

An expansion valve is usually rated on the basis that liquid only (no vapor) will reach its inlet. Where the manufacturer cannot supply valve ratings for mixtures of liquid and vapor, a formula[1, 2] of the following type will give results with a sufficient degree of accuracy for most applications:

$$T = C\sqrt{Pd}$$

Therefore, the following formula may be used to compute the capacity of a valve for any required pressure difference and fluid density if the valve capacity is known for any other pressure difference and liquid density.

$$\frac{T}{T_1} = \sqrt{\frac{Pd}{P_1 d_1}} \quad \text{.....[1]}$$

where

P_1 = Pressure difference for which valve is rated,

P = Actual pressure difference under which valve must operate,

d_1 = Liquid density for which valve is rated,

d = Actual density of mixture of "Freon" liquid and vapor delivered to the valve,

T_1 = Known tonnage rating of the valve for a given pressure difference and liquid density,

T = Actual tonnage rating of valve at the actual pressure difference and density.

If the liquid density for which a particular valve has been rated is known, the actual value should be used in Formula 1. If it is not known, the value of 80 lb per cu ft may be used. Considering other possible sources of error, this value will be sufficiently accurate.

Example 5: Taking the conditions of Examples 1 and 3, find the actual capacity of an expansion valve rated at 15 tons for a 90 lb pressure difference and "Freon" liquid having a density of 80 lb per cu ft.

Solution: From Example 3,

Actual pressure difference = 59 lb.

Table 1—Saturation Pressures of "Freon"

(*From A.S.R.E. Data*)

Temperature Deg Fahr	Gage Pressure Lb per Sq In.	Temperature Deg Fahr	Gage Pressure Lb per Sq In.
30	28.5	80	84.1
32	30.1	82	87.0
34	31.7	84	90.1
36	33.4	86	93.2
38	35.2	88	96.4
40	37.0	90	99.6
42	38.8	92	103.0
44	40.7	94	106.3
46	42.7	96	109.8
48	44.7	98	113.3
50	46.7	100	116.9
52	48.8	102	120.6
54	50.9	104	124.3
56	53.1	106	128.1
58	55.4	108	132.1
60	57.7	110	136.0
62	60.1	112	140.1
64	62.5	114	144.2
66	65.0	116	148.4
68	67.5	118	152.7
70	70.1	120	157.1
72	72.8	122	161.5
74	75.5	124	166.1
76	78.3	126	170.7
78	81.2	128	175.4

[1] The Measurement of Expansion Valve Capacity, by D. D. Wile, *Refrigerating Engineering*, August, 1935, p. 79.

[2] "Freon-12" Thermal Expansion Valve Requirements, by J. G. Bergdoll, *Refrigerating Engineering*, December, 1936, p. 405.

From Example 4 (for the conditions of Example 3) density of "Freon" mixture = 43 lb.
Substituting in Formula 1,

$$\frac{T}{15} = \sqrt{\frac{59 \times 43}{90 \times 80}} = \sqrt{0.352} = 0.593$$

$$T = 8.9 \text{ tons}$$

As a rule, the problem usually encountered is the reverse of the preceding one. It is usually necessary to select an expansion valve for a given set of operating conditions. Formula 1 can be used for this purpose also, as illustrated by the following example.

Example 6: A refrigerating system operating under the conditions of Examples 1 and 3 is to have a capacity of 50 tons. Find the required total capacity of the expansion valves (if several are used) under their standard rated condition. The standard rating condition is, say, a pressure difference of 85 lb with liquid "Freon" at a density of 80 lb per cu ft.

Solution: From Example 3, actual pressure difference = 59 lb per sq in.

From Example 4, actual density = 43 lb per cu ft.
Substituting in Formula 1,

$$\frac{50}{T_1} = \sqrt{\frac{59 \times 43}{85 \times 80}} = 0.60$$

$$T_1 = \frac{50}{0.60} = 83.3 \text{ tons}$$

Thus for the conditions of this example, expansion valves having a total rated capacity of 83.3 tons would be required.

It is possible to locate an evaporator a considerable distance above the receiver. However, it is best to limit the vertical distance, because the capacity of available expansion valves may be reduced to such an extent that several large valves may be required for a relatively small load. For large loads, it may even prove impractical to install the required number of large valves. This point can only be settled for a specific installation by referring to manufacturers' capacity data for expansion valves.

[*To be concluded. Part 3 will describe and illustrate the various methods of lessening the amount of liquid refrigerant that flashes in vertical liquid pipe lines, and discuss the effect on selection of expansion valves when evaporating coils are below the condenser, such as with evaporative condensers installed on the roof, a type of job which is coming in to common use.*]

Office Building Air Conditioning Pays Health Dividends

HOW air conditioning in Tribune Tower, Chicago, has armed *Tribune* employees against colds, influenza, and kindred infections was disclosed last month. The survey reveals that air conditioning pays solid health dividends.

The survey was begun in May, 1933, exactly twelve months before air conditioning was installed in the Tower. At that time there was an average of 2,150 persons on the *Tribune* pay roll each month.

During the twelve months from May, 1933, to May, 1934, a total of 475 persons were compelled by various respiratory diseases to miss one or more days of work. These 475 employees missed a total of 2,228 days of work because of illnesses, an average of about five days each.

On May 10, 1934, the air conditioning system for the entire Tribune Tower was placed in operation. The results became apparent immediately.

Despite the fact that for the year of May, 1934, through April, 1935, the *Tribune* employed an average of 2,500 persons a month, only 345 became ill enough to miss a day of work, and these persons missed only 1,290 days, an average of slightly more than three and a half days a person.

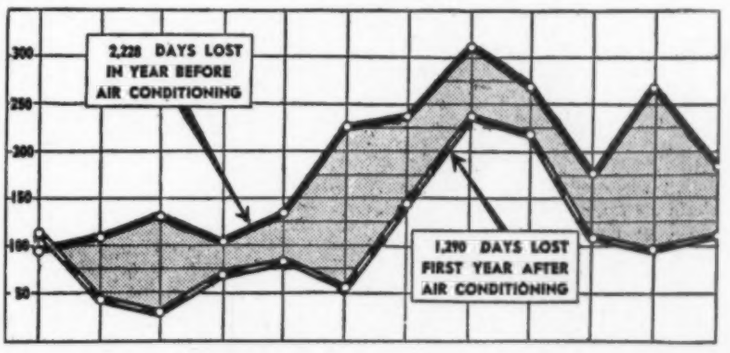
Stated in another way, 22 per cent of the total number of employees during the year May 1, 1933, through April 30, 1934, became ill enough to remain away from their jobs one or more days. For the subsequent twelve months the percentage dropped to 13 per cent of the total number of employees.

During the remainder of 1935, from May 1 through December 31, only 204 persons missed days of work, although the average number of employees in those months was 2,800.

In the epidemic year 1936, when influenza was rampant, only 571 *Tribune* employees out of an average of 3,000 on the pay roll were forced to stay home because of respiratory illnesses.

Dr. Irving S. Cutter, dean of the Northwestern University medical school and the *Tribune* health adviser, said that the excellent health record of *Tribune* workers during the last two years was undoubtedly due in a large measure to air conditioning at the Tower.

The solid line indicates the days lost by *Tribune* employees before installation of the air conditioning system. The broken line below illustrates the days lost after the system was placed in use. The shaded portion between emphasizes decrease in loss of time



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Boston's Rice Building

Modernized with Air Conditioning

By **Frank A. Merrill*** **[as told to Francis A. Westbrook]**

NO. 10 High Street, Boston, Mass., is the address of one of the most modern public buildings in New England, for in its marble corridors and 325 private offices are now enjoyed the advantages of a complete system of air conditioning installed last summer, which probably makes the 11 story, 20 year old Rice building America's largest privately owned, completely air conditioned building with offices for public rental.

As the typical floor plan is roughly in the form of an H, it was deemed advisable to lay out the air distribution in two zones, one for each wing. The 210 ton "Freon" refrigerating plant is in the basement; chilled water is pumped to eight apparatus rooms, four in each wing, on the 2nd, 5th, 8th and roof levels. The ducts in the offices are furred and plastered, this not being a new building, and separate air inlets in each office provide for individual manually operated volume control. The amount of cool air supplied to the various parts of the building in summer is automatically controlled in accordance with the sun heat load.

The equipment in each of the apparatus rooms includes a bank of cooling coils on the suction side of the fan, which is also installed in the apparatus room. Of course each bank of cooling coils has sufficient surface to handle the amount of air required for the space allotted to it; this of necessity varies somewhat for the different apparatus rooms as it was not practicable to divide the space in exactly even parts. The outside air ducts are each equipped with a preheater to temper the outside air during the winter. There is also a nest of spray nozzles with each bank of coils for humidifying the air in winter.

A proper amount of outside air is mixed with the returned air in the apparatus rooms and passes over the cooling coils. After passing through the duct system to the offices and picking up its heat load, the air goes into the corridors through the transoms which are kept slightly opened. The corridors act as return air ducts and convey the air to the return grilles which conduct it to the plenum return space in the apparatus rooms.

Outlet grilles are of the aspirator type and especially designed for distributing conditioned air in small offices. Behind each of these grilles, and between it and the supply duct, there is a manually operated volume control damper so that each occupant may control the volume of air as he sees fit. It is felt that this feature, together with the automatic volume control associated with the shifting sun heat load, gives the greatest practicable flexibility to the air distributing system.

Automatic control is accomplished by means of two thermostats governing the air supplied by each fan room. One thermostat compensates for morning sun and the other for the afternoon sun.

These thermostats control the operation of automatic volume dampers in the ducts which feed air to each section of the zone. Thus the thermostat controlling the eastern exposure will permit the circulation of additional cooled air on that side of each wing in the morning by partially closing the damper in the duct supplying the westerly exposure, and permitting more cooled air to flow to the warm side of the building and less to the shaded side. In the afternoon, as the sun shifts to the western side of each wing, the reverse operation takes place. The controls are actuated by a sub-master thermostat in each apparatus room and a master thermostat on the outside wall shielded from direct sunlight.

The circulation of chilled water to the cooling coils is also automatically controlled. That is, the capacity of the compressors is automatically varied from 100 to 50 per cent by a thermostat which is submerged in the water cooler. This actuates a solenoid valve which controls the capacity of the compressor within these limits. The compressors also are equipped for 25 per cent reduction in capacity by means of manually operated valves.

The controllers for the motors in the apparatus rooms on the various floors operate on remote control from the basement engine room where the refrigerating machinery is situated. The outside air duct in each apparatus room has an automatic damper which closes when the fan motor is shut down, as does the automatic steam valve feeding the heating coils and the water valve on the line supplying the humidifying spray nozzles.

The system has been designed to heat in winter 24,300 cfm of outside air to 75 deg with an outside temperature of 0 deg with a relative humidity of 30 per cent when the inside temperature is not above 75 deg. In summer the system is designed to maintain an 80 deg temperature with a relative humidity not over 50 per cent, with the outside dry bulb temperature 95 deg and the outside entering wet bulb temperature 75 deg; this being subject to the additional condition that inside sources of heat arise from 1320 people, 44 kw of lighting, 30 motor hp, as in the apparatus rooms, and the natural inleakage of warm air which cannot reasonably be avoided.

*Office of Hollis French, Consulting Engineers.

Further operating conditions are that the windows and other openings, as doors, be kept closed normally, and that shades or venetian blinds will be used at the windows. The noise level within the conditioned area has been guaranteed not to be increased by more than 5 decibels above normal room noise level when measured 10 ft from the supply air grilles.

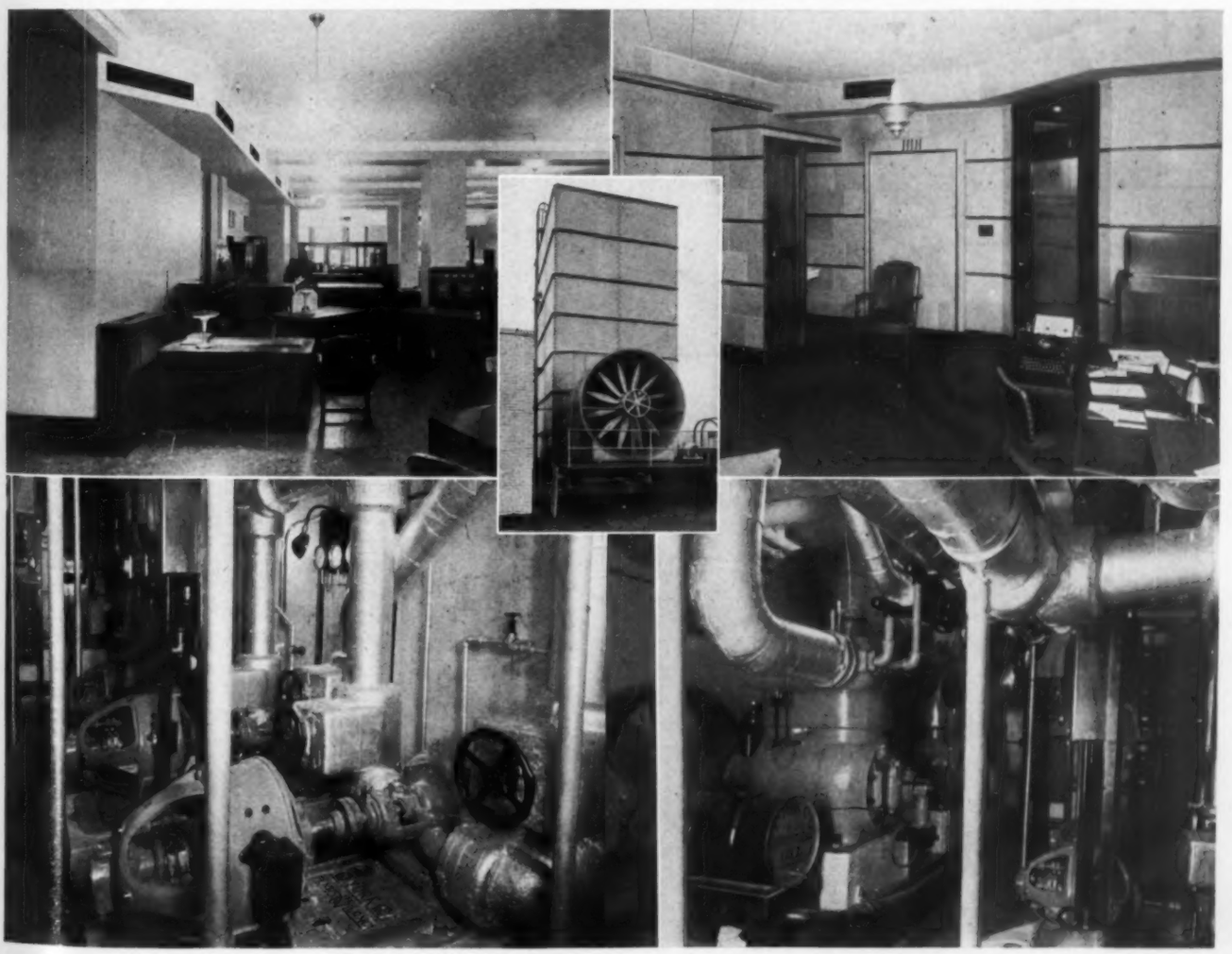
A study of the log sheets taken last summer revealed the interesting fact that on several days when the outside temperature did not exceed 80 deg one compressor with a capacity of 105 tons was running at full load all day. There were also two or three days when the temperature did not exceed 83 deg when both compressors were running at full load in order to produce the desired conditions in the offices. This is apparent when it is considered that the transfer of heat through the building is only about 20 per cent of the total heat load, the balance coming from the shifting sun load and outside humidity conditions.

That it is good business to air condition an existing building is shown by the fact that when the system was installed 58,000 sq ft of the available space was rented and that within a few months this was increased to 68,000 sq ft in the face of severe competition from newer buildings. The total rentable space in the building is 93,000 sq ft. The overall cost of the air conditioning system was $125,000 completely installed and placed in operation, which gives a net cost of approximately $1.35 per sq ft of rentable area, which compares favorably with the generally accepted figure for air conditioning a large office building of from $1.25 to $1.45 per sq ft rentable area.

Easy to Remove Studs

An old idea, but new to many and finding frequent use, is the easy removal of bolts and studs that have broken off flush with the surface or below the surface of the metal into which they have been screwed. Weld a stem onto the old stub and bring it up to a flat shape to which a wrench can be applied. During the welding process the stub of the old bolt expands against the surrounding metal and when it cools it becomes slightly smaller than it was before heating. This action together with the ease of applying the wrench by means of the welded-on stub makes it fairly easy to remove broken off bolts. This method has been successfully used down to 3/16 in. stove bolts.—*Oxy-Acetylene Tips.*

Conditioned air grilles in a large office (left) and a small office (right). At the lower left is a view of the pumps and at the lower right may be seen the compressors. The center view shows the cooling tower



Heat Recovery System to Reduce Laundry Steam Consumption

By V. F. Alley*

THE Philadelphia Electric Co. recently secured a three year contract to supply district steam to the largest laundry in the city for their entire process, hot water and heating requirements. The customer will use 30,000,000 lb of steam at an annual cost of $24,000.

Several interesting facts encountered in the plant are reported, among them being that the boilers were in good operating condition with an average evaporation rate of 9.16 lb of steam per lb of coal and that they were generating 50,000,000 lb of steam annually when 30,000,000, properly used, would do a better job.

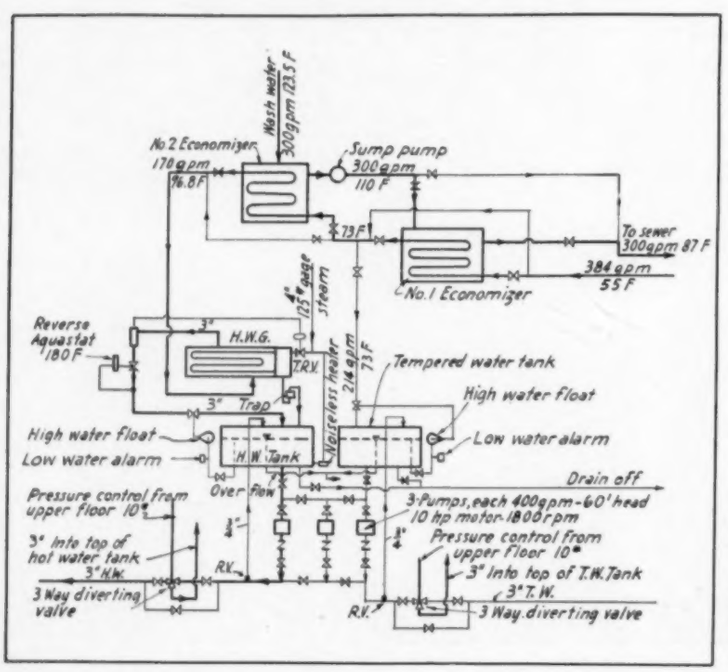
An adequate hot water supply was not available and it was necessary to use live steam jets in the wash tubs to raise the wash water to the required temperature, mangles when not in use were not turned off, the heat reclaiming system was inefficient and the process and hot water lines were generally in bad condition. It was apparent that a process and hot water load of this magnitude operating under such conditions offered an unusual opportunity for improvement. With this thought in mind, a thorough study was made of all operations in the plant.

Recommendations were made which will insure efficient utilization of district steam, the most important being the new hot water and heat recovery system shown in the diagram by which incoming city water at 55 deg will be raised to 96.8 deg. in primary and secondary economizers.

This result is accomplished by utilizing the heat in the water from the wash room which returns at 123.5 deg and after passing through the heat reclaimers will discharge to the sewer at 87 deg. A 7,500 gal hot water and a 6,000 gal tempered water storage tank with the necessary pumps and control equipment complete the system.

*Manager, Industrial Sales, Philadelphia Electric Co., Philadelphia, Pa., in a recent issue of the *Bulletin* of the *National District Heating Association*.

Heat recovery system for laundry. *H.W.G.*: Hot water generator. *R.V.*: Relief valve. *T.W.*: Tempered water.



Guest Room Air Conditioning Expected to Create Hotel Business

Four downtown St. Louis hotels will have air conditioned guest rooms this summer. Will competition cancel the benefits of these installations, so that each hotel will wind up with approximately the same amount of summer business as before?

"No," answers Manager Charles Heiss of the Mayfair and Lennox hotels, which were recently completely air conditioned. "During the hot summer months, we believe that a large number of St. Louisans living in West End residential districts will move into hotels where they can enjoy a refreshing sleep in an air conditioned room. Thus a new class of local clients will be created where none existed before.

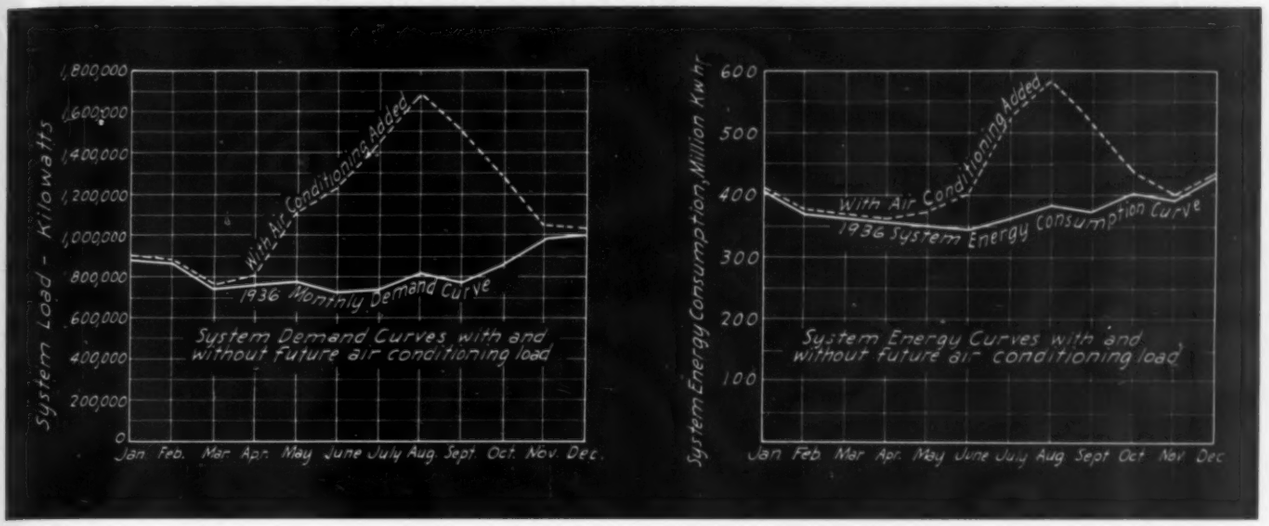
"Furthermore, we feel that there is a growing tendency in this part of the country for people living in smaller cities to spend their week-ends and holidays in nearby large cities where they can find air conditioned hotel rooms, as well as cooled shops, theaters and restaurants. This type of week-end visitor will travel 100 miles or more to reach the nearest large city.

"At present we are making a survey to determine the extent of this tendency. Although it is not yet completed, first results of the study seem to show that air conditioned metropolitan hotels can successfully compete with resort hotels for out-of-town trade, even in cases where the resort is nearer to the outlying city."

The new installations for the Mayfair and Lennox serve 35 floors of guest rooms, 700 rooms in all. "Freon" refrigerating machines in the basements supply cooled water to conditioning units, each of which handles three room floors in the case of the Mayfair and two floors in that of the Lennox.

Approximately 190 tons of refrigeration is produced in each hotel. Total horsepower of all motors, including those which drive the fans, is 307 at the Mayfair and 402 at the Lennox.

Cost per room of the installation was approximately $300, according to Harry Maurer, chief engineer of the two hotels, and the cost of operation is expected to be about 50c a day.



How Much Air Conditioning?

By **William P. Rock***

A Survey of the Ultimate Market for Summer Air Conditioning in a Large City, and Its Possible Effect on the Power Company. . . . . There Are Now 1466 Installations of "Electric" Air Conditioning in Chicago and May Sometime Be 200,000 or More

DETAILS of the average operating characteristics and power usage of all important types of residential and commercial air conditioning installations in Chicago were given in the April issue. The figures for this survey were collected for each month in 1936 and were obtained by averaging actual electric meter readings.

The present article gives further facts from the same survey, first, with respect to the apparent ultimate saturation of the air conditioning market when viewed in the light of today's knowledge, and, second, with respect to the possible future effect on the power company if all the ultimate saturation is accomplished.

To arrive at the ultimate growth and expansion of air conditioning, an economic approach is chosen. An estimate of the families and businesses financially able to purchase the necessary equipment is considered here as a reasonable estimate of the saturated market. There is no time element in the survey and it should not be taken as a prophecy, for no one ever knows how soon, or how completely, an ultimate saturation is likely to be reached.

An article in the January issue gave details of making an economic survey for the purpose of estimating the ultimate air conditioning market. As described therein, the city was divided into 48 sections, and each section was assigned an economic rating in accordance with the family buying power of that section. All businesses located within the limits of the highest rated economic sections are considered as those likely to be air conditioned in the future. Surveys have been made of rentals or equivalent rentals paid by each family. This study assumes that those families paying the highest rents or owning homes that represent the highest equivalent rents will eventually enjoy the benefits and comforts of air conditioning.

The results of the economic survey of 13 classifications of air conditioning are tabulated in Table 1 under the heading "Estimated Future Installations." In order to show the "present" along with the "future," the number of present installation in each classification is indicated for the purpose of comparison.

Table 1—Estimated Number of Future Air Conditioning Installations Using Refrigeration for Summer Cooling

Type of Installation	Present Installations	Estimated Future Installations
Residential		
Room Units	87	106,000
Central Plants	29	106,000
Commercial		
Beauty Shops	15	2,051
Doctors and Dentists	24	3,650
Funeral Parlors	33	495
Hotels	28	235
Offices, General	144	4,144
Restaurants	208	2,327
Stores		
Candy	31	675
Clothing	48	984
Drug	34	1,150
Shoe	25	250
Theaters	149	256

*Air Conditioning Div., Commonwealth Edison Co.

The possible effect of the fulfilled air conditioning market on the power company is now rather easily obtainable. Average monthly electrical operating characteristics and usage for each of the 13 classifications listed in Table 1 were tabulated and charted last month. They were the results of averaging monthly meter readings on many installations and showed the varying usage of air conditioning equipment throughout the year 1936. These average monthly electrical characteristics of each class of installation when multiplied by the estimated ultimate future number of air conditioned establishments of that classification will show the ultimate possible electrical usage of that classification. By summing up the future monthly usage of all classifications, the total usage of all 13 classifications is obtained for each month.

Table 2 shows these total monthly usages. It also shows the 1936 monthly electrical characteristics of the Commonwealth Edison Co. system on which will be placed this estimated air conditioning load of the future. The charts obtained by plotting the monthly figures of both the present system characteristics and the future air conditioning characteristics picture rather forcefully the possible increase in electrical usage to be brought about by future summer cooling in connection with air conditioning.

Table 2—1936 System Operation Compared with Possible Ultimate Load to Be Added by Growth of Summer Air Conditioning

Month	1936 System Kw	Estimated Air Conditioning Kw To Be Added	1936 System Kwhr In 1000's	Estimated Air Conditioning Kwhr In 1000's
Jan.	883,000	14,300	405,109	3,070
Feb.	871,000	12,200	377,977	2,915
March	737,000	12,149	368,386	2,300
April	768,000	38,705	357,207	5,711
May	783,000	386,840	347,963	32,242
June	725,000	507,227	345,008	57,679
July	737,000	713,886	369,293	158,413
Aug.	816,000	870,430	380,135	209,053
Sept.	778,000	736,100	373,222	138,335
Oct.	868,000	427,050	401,878	37,060
Nov.	986,000	67,220	393,097	5,563
Dec.	996,950	40,250	434,473	3,600

This survey of the ultimate market for summer air conditioning in a large city and its effect on the power company, has included 13 classifications of business and residences. The estimated ultimate market of 200,000 or more installations has not included any industrial businesses such as printing, baking and candy making, or certain classes of commercial businesses such as amusements, banks, barber shops, churches, clubs, hospitals and food stores.

Five Air Conditioning Plants for World's Fair Building

By H. G. Schaefer

TWO years hence New York City will sponsor the World's Fair of 1939. On the basis of the present geometric ratio of acceleration of the exploitation of scientific developments, it is questionable whether even a contemporary Jules Verne could foretell, with any reasonable accuracy, what marvels of industry, engineering, and the arts and sciences will be laid before the world at this event.

The selection of the site of the Exposition was made several years ago. It lies in the basin of the once tidal Flushing River, which flows northward through the western end of Long Island, and empties into Long Island Sound. The stream will be practically obliterated by the construction of a tidal control gate, but will serve as an outlet for the two connected spring-fed lagoons that are a feature of the development.

The site lies within the boundaries of Greater New York, and is within fourteen minutes, by rail, of Times Square. It borders the new parkway connecting with the Tri-Borough Bridge, and can be conveniently reached by the three subway systems, as well as by the Long Island Railroad. Grading and landscaping of the 1216 acre tract is now in progress, and the planning of the building disposition is practically complete.

In order to facilitate the construction of the Fair buildings and the general preparations for the opening of the Exposition, as well as to provide operating headquarters, a building to house the administrative and executive organizations is now under construction. This, the initial structure of the city that will shortly be reared on the site, is two stories high and approximately 458 ft long by 180 ft wide, with an inside court, and will provide 57,000 sq ft of net floor area. It is located at the intersection of the Long Island Railroad line with Grand Central Parkway, one of the vehicular traffic arteries recently completed in the extensive program of the municipal park department.

The heating, ventilating and air conditioning equipment, now in process of installation by Baker, Smith & Co., Inc., in the administration building includes four 60 ton direct expansion "Freon" plants, two of which will serve the general offices, one the executive offices, and one the cafeteria. A 7½ ton unit will serve the model exhibition room. Both summer and winter conditioning will be achieved by pneumatic thermostatic and hygrostatic control, and distribution of the 60,000 cfm of conditioned air will be aided by sound-proofed ductwork and directional high velocity grilles.

The heating will be done by means of a two pipe vacuum system which will distribute the steam generated by gas-fired boilers to cast iron radiators and convectors, which, in the case of the executive offices will be controlled by self contained thermostatic radiator valves. The direct heating load will be approximately 13,500 sq ft.

The Strength and Design of Covers and Flanges for Pressure Vessels and Piping

[Part 7]

By T. McLean Jasper*, H. Gregersen*, and A. M. Zoellner*

THE construction of a vessel or pipe line for high pressure service involves a balance in the design at every point so that the safety of such equipment can be assured. Previous investigations have established the necessity for and the required amount of reinforcement for openings in vessel walls and for the proper shapes of heads of vessels. The present investigation aims to go into the question of covers and flanges of openings.

This is the concluding installment of this paper, the first part of which was published in the November, 1936, issue.

APPENDIX 5—HISTORY OF FLANGE THEORIES

Fig. 34—Reverse loading of flange

Ring Flanges

Waters' and Holmberg-Axelson's formulas for ring flanges are based on the general plate theory, and Timoshenko's formula is based on the theory of deformation symmetrical about an axis. Since Waters' and Holmberg - Axelson's formulas are based on the same theory, there cannot possibly be any essential difference between the two formulas. In the following discussions it will be shown how Holmberg-Axelson's formula for ring flanges can be converted into Waters' formula for ring flanges:

The general form of Holmberg-Axelson's formula for ring flanges (Equation 6) is:

$$S = \frac{3}{2\pi} \frac{W}{T^2 (D^2 - d^2)} \left[(1-\mu)(b^2 - g^2) + 2(1+\mu) D^2 \log_e \frac{b}{g} \right]$$

Removing G to the inner edge, and W to the outer edge of the ring flange (Fig. 3, p. 606, November, 1936, issue) g and b become equal to d and D respectively which inserted in the above formula gives:

$$S = \frac{W}{T^2} \left[\frac{3(1-\mu)}{2\pi} + \frac{3(1+\mu)}{\pi} \frac{D^2}{D^2 - d^2} \log_e \frac{D}{d} \right] \ldots\ldots [49]$$

Equation 49 multiplied and divided by $(D\text{-}d)$:

$$S = \frac{W}{T^2 (D-d)} \left[\frac{3(1+\mu)}{\pi} \frac{D^2 (D-d)}{D^2 - d^2} \log_e \frac{D}{d} + \frac{3(1+\mu)}{2\pi} (D-d) \right] \ldots\ldots [50]$$

List of Symbols Used

W—Total bolt load
G—Gasket load
D—Outside diameter of cover, ring, or flange
T—Thickness of cover, ring, or flange.
S—Stress
S_r—Modulus of rupture
$M = Wa$—External moment applied or assumed in design
M_e—Moment per unit length of circumference
Q—Force
Q_e—Shearing force per unit length of circumference
M_{test}—Moment applied in test samples at failure
S_{test}—Stress obtained in test samples at failure
b—Bolt circle diameter
g—Gasket diameter
d—Inside diameter of flange or ring opening
a—Distance between gasket and bolt circle
a_e—Distance from bolt circle to center of hub thickness in a hubbed flange
d_m—Mean diameter of hub in a hubbed flange
t—Thickness of hub in a hubbed flange
μ—Poisson's ratio

Substituting $D - d = 2a$ inside of the brackets in Formula 50:

$$S = \frac{3}{\pi} \frac{Wa}{T^2 (D-d)} \left[2(1+\mu) \frac{D^2 \log_e \frac{D}{d}}{D^2 - d^2} + (1-\mu) \right]$$

which is identical with the general form of Waters' formula for ring flanges (Equation 7).

Waters' formula has one variable less than Holmberg-Axelson's formula, is accurate enough for all practical dimensions of ring flanges, and has the advantage of expressing the maximum stress as a function of the external moment Wa.

Hubbed Flanges

All modern formulas for hubbed flanges are based on the theory of bending of bars on elastic foundation and the general plate theory, or the theory of deformation symmetrical about an axis. One of the first attempts to develop a formula based on these methods was made by Waters. This work, however, was based on the formula for the tangential stress at the inner edge of the ring part of the flange, which generally, is far smaller than the maximum stress which occurs at the joint between ring and hub. Thus the application of this formula will result in hubbed flanges having considerably lower strength than indicated by the calculations. From the tests reported here (made with hubbed flanges) we know that the strength increases considerably with increased fillet, and thus the application of the above mentioned formula would be justified, if a fillet could be applied large enough to remove

*Research Engineer, A. O. Smith Corp.
Illustrations reproduced by courtesy of the A. O. Smith Corp.
This paper was presented at the annual meeting of the *American Society of Mechanical Engineers*, December, 1934, New York, N. Y., but has not heretofore been available in published form.

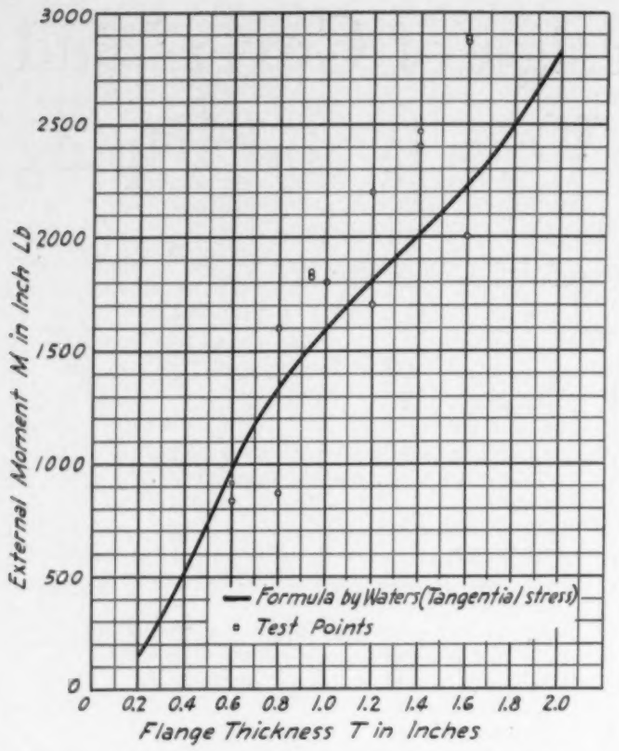


Fig. 35—Test points and theoretical curve calculated from Waters' formula showing variation of external moment with flange thickness for small hubbed flanges subjected to reverse loading

(Note: Waters' formula does not apply to the point at which maximum stresses occur in a hubbed flange).

the maximum stress from the joint between ring and hub to the inner edge of the ring. The authors decided to investigate the possibility of such a removal of the maximum stress by tests. Since the inner edge of the ring section is stressed in compression, and the compressive strength of the test material is four times the tensile strength, the general test arrangement shown in Fig. 20 could not be used for this purpose. It was decided to subject the flanges to a reverse test as shown in Fig. 34, whereby all compressive stresses produced in the general test arrangement shown in Fig. 20 will be converted into tensile stresses and vice versa. The test results and their evaluation are shown in Fig. 35 together with the theoretical curve calculated from Waters' formula for flanges with long hubs:

$$S = \frac{5.72\,Wa}{d_m\,(K-1)\,T^2}\left[\frac{K^2\log_{10}K}{K^2-1} + 0.117 - \frac{K^2+1}{K^2-1}\left(\frac{3.33\,K^2\log_{10}K + 0.398\,(K^2-1)}{4.33\,K^2+2.33+(K^2-1)\,T^3\,\frac{1.56}{\sqrt{d_m\,t^3}}}\right)\right] \quad \ldots\ldots [51]$$

where $K = \frac{D}{d_m}$ ..... [52]

and S is the tangential stress at the inner edge of the ring part of the hubbed flange. From Fig. 35 it can be seen that the conformity between the test results and the theoretical curve is within technical accuracy.

It should be pointed out, however, that Waters' formula does not give the stresses at the point of maximum stress until the fillet is increased at the junction between hub and flange to considerably above a practical value for the fillet.

Hubbed Flanges—Reverse Loading—Variation of Flange Thickness

(*Fig. 35 has been plotted from the values given in this table.*)

Flange Dimensions		Average Tensile Strength of Material:	$S_{Test} = \frac{S_r\ Aver.}{1.2}$
$D = 10.1$ In.	$b = 9.4$ In.	$S = 318$ Lb per Sq In.	$M = \frac{318}{S_{Test}} M_{Test}$
$d = 5.05$ In.	$g_1 = d_m = 5.7$ In.		
$t = 0.65$ In.	$a = 1.85$ In.		
$R = \frac{1}{2}$ In.	$g = 5.57$ In.		
$H = 4.5$ In.			

T	M_{Test}	S_r Aver.	S_{Test}	M
0.6	948	400	333	905
0.6	1076	498	415	825
0.8	2210	528	440	1600
0.8	1115	494	412	861
1.0	1670	354	295	1800
1.2	2270	394	328	2200
1.2	1930	433	361	1700
1.4	2620	418	348	2400
1.4	3170	491	409	2460
1.6	2060	394	328	2000
0.933	1690	356	297	1810
0.933	2390	501	417	1820
1.6	3860	533	444	2765
1.6	3870	522	435	2830

(All numerical values in in.-lb system.)

Fig. 36—Typical fracture of flange subjected to reverse loading

Fig. 36 shows the typical fracture of a flange subjected to reverse loading.

A comparison between Fig. 9 or Fig. 11 and Fig. 35 shows that the type of curve obtained from Holmberg-Axelson's and Timoshenko's formulas for the maximum stress in hubbed flanges is considerably different from the curve obtained from Waters' formula for the tangential stress at the inner edge of the ring part. Their difference becomes greater the larger the flanges are and for very large flanges having an outside diameter of 75 in. and more, Formula 34 by Waters results in relations between flange thickness and strength impossible for practical purposes.

[*Concluded*]

There is little doubt that a general air conditioning business is an engineering business. An organization must consist of engineers but there must be additional specialized talents for designing, selling, installing, servicing and management. The ideal combinations are not easy to find. The firm so fortunate as to have the required specialized engineering talent and a sufficient working capital may be said to be truly organized for business.—H. W. Reding, Danforth Co.

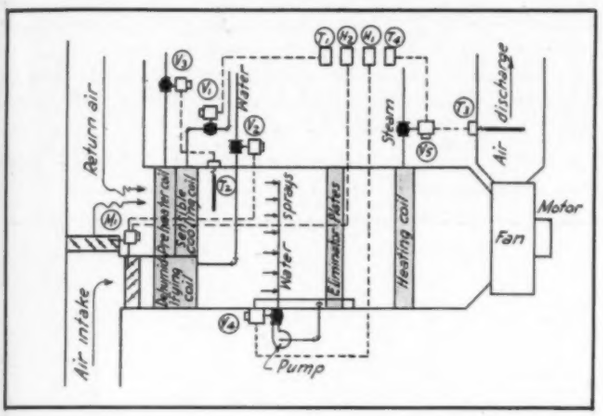
CONTROLS

By C. L. Ringquist*

Reference Sheet 9

Heating, Humidifying, Ventilating, Dehumidifying, and Cooling Systems

One of the many arrangements of this type:



Cycle—For heating, motor or diaphragm operated valve V_3 governed by insertion thermostat T_2 controls the steam or hot water supplied to the preheating coil in the air stream mixture. The damper in front of the dehumidifying coil is closed during the heating cycle, thus putting the mixture of outside and return air through the preheating coil and maintaining a constant temperature in the spray chamber.

Thermostat T_4 controls motor or diaphragm operated modulating valve V_5 which in turn controls the supply of steam or hot water to the reheating coil in the air stream after the sprays. This valve is also interconnected to insertion temperature controller T_3 in the fan discharge. T_3 is set so that the delivery temperature from the fan will never go below a definite minimum setting.

For humidity control in the heating cycle, room humidistat H_1 controls the operation of motorized or diaphragm valve V_4 (of the modulating type) in the spray water line discharge from the pump. This valve controls the amount of water circulated through the spray, and when humidity is not required, it will close entirely and with the aid of an auxiliary switch on the valve, will shut down the pump.

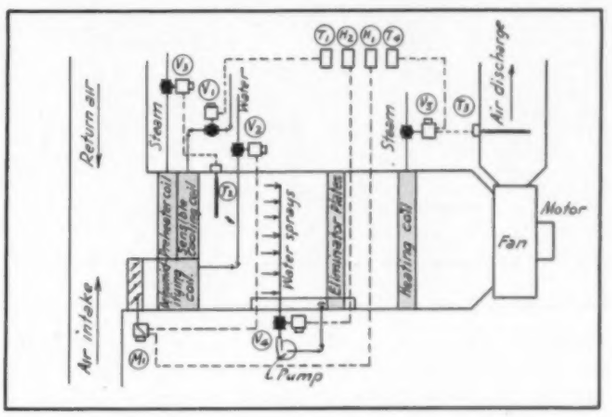
For the cooling cycle, room thermostat T_1 in the conditioned space governs the motorized or diaphragm valve V_1 (which is of the modulating type), and controls the supply of cold water or refrigerant to the sensible cooling coil. Whenever a drop is required in the dry bulb temperature of the air, this thermostat will control the amount of refrigerant flowing through the sensible coil and thus the room temperature.

For humidity control during the cooling season, room humidistat H_2 governs the operation of damper controller M_1 (which is of the modulating type) and which in turn controls the operation of the face and shunt dampers in the mixing chamber. M_1 also is equipped with an auxiliary switch so that when no air is passing through the dehumidifying coil, V_2 will be entirely closed. This valve is also of the modulating type.

The operation of this cycle is then such that when the humidity of the room is too high, more air will be passed through the dehumidifying coil. As the humidity in the room drops, the face damper on the dehumidifying coil will start to close and circulate more of the air through the sensible cooling coil.

*Air Conditioning Department, The Trane Co.

The following system is similar to the one just described with the exception that the dehumidifying and sensible cooling coils have been arranged in somewhat different manner:



Cycle—The controls for heating consist essentially of insertion thermostat T_2 in back of the preheating coil which operates or controls motorized or diaphragm valve V_3 (of the modulating type) which in turn controls the steam or hot water supply to the preheating coil. During the heating cycle, the motorized damper in front of the dehumidifying coil is closed at all times to prevent freezing of the sprays.

Room humidistat H_2 controls the operation of motorized or diaphragm valve V_4 (of the modulating type) in the pump discharge line. V_4 controls the amount of water circulated to the sprays, thus governing the humidity added to the air stream.

Room thermostat T_4 controls the temperature in the room through modulating type motorized or diaphragm valve V_5 governing the amount of steam or hot water supplied to the reheating coil. V_5 is also interconnected to duct thermostat T_3 in the fan discharge, so that the air temperature cannot fall below a specified minimum value.

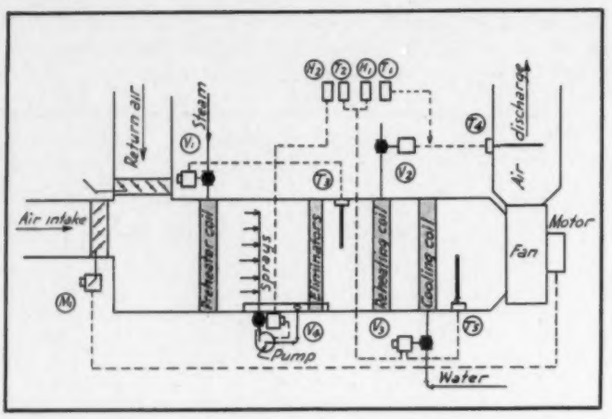
For summer operation, room thermostat T_1 controls modulating type motorized or diaphragm valve V_1 in the refrigerant or water supply line to the sensible cooling coil. Modulation of V_1 controls the amount of cooling supplied by this coil and, therefore, the room temperature.

For humidity control during the cooling cycle, humidistat H_1 in the conditioned area controls the operation of modulating type damper motor M_1 which governs the amount of air circulated through the dehumidifying coil. M_1 is also equipped with an auxiliary switch so that when the dampers are in the closed position, V_2 will shut off the refrigerant or chilled water supply to the dehumidifying coil.

Another arrangement for a system of this type:

Cycle—For heating, modulating type motorized or diaphragm valve V_1 in the steam line to the preheater coil is controlled by insertion thermostat T_3 located after the sprays and set to maintain a constant temperature from the spray chamber. Modulating type motorized or diaphragm valve V_2 in the steam supply to the reheating coil controls the room temperature from room thermostat T_1 while modulating the steam supply to the reheating coil. V_2 is also interconnected to duct thermostat T_4 to control the discharge air at a definite minimum temperature; T_4 acts as a low limit control.

Damper controller M_1 (of the two position type) is set with a spring return so that when the fan motor is not in operation, the outside air intake damper is closed. The return air damper is set in a position so that a definite amount of return air and a definite amount of outside air is mixed in the mixing chamber in front of the preheating coil. Whenever the fan motor is started, the outside air intake is opened by means of M_1 to a definite fixed position.



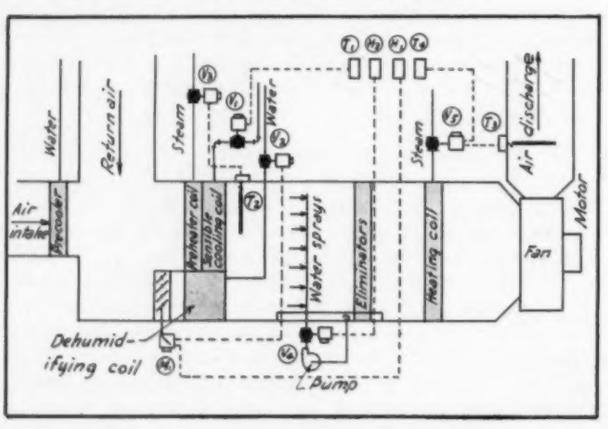
The humidity control for winter operation consists of humidistat H_2 in the conditioned area controlling the amount of water circulated by the pump to the sprays. Humidistat H_2 controls motorized or diaphragm valve V_4 (which is of the modulating type) in the discharge water line from the pump. V_4 is also equipped with an auxiliary switch so that when it is entirely closed, the pump motor is shut down.

During the cooling cycle, thermostat T_2 in the conditioned area controls motorized or diaphragm valve V_3 (of the modulating type) in the return line from the cooling coils. By thus governing the amount of refrigerant or water circulated through the cooling coils, the room temperature is definitely maintained.

Room humidistat H_1 is interconnected to V_3 so that in cases where temperature of the conditioned space is down and the humidity is not as low as required, the humidistat will assume control of the supply of refrigerant to the cooling coil. V_3 is also equipped with an auxiliary contact controlled by means of temperature controller T_3 in the space between the cooling coils and the suction of the fan. T_3 assumes control of V_3 whenever thermostat T_2 and humidistat H_1 are entirely out of operation, to maintain a definite temperature of air entering the fan and prevent extreme variations in the temperature of the air introduced in the conditioned space.

Damper motor M_1 is also operated by means of the fan motor during summer operation.

This system operates the same as the second example, with the exception that a precooling coil is placed in the outside air intake in order to reduce the size of the cooling coils:

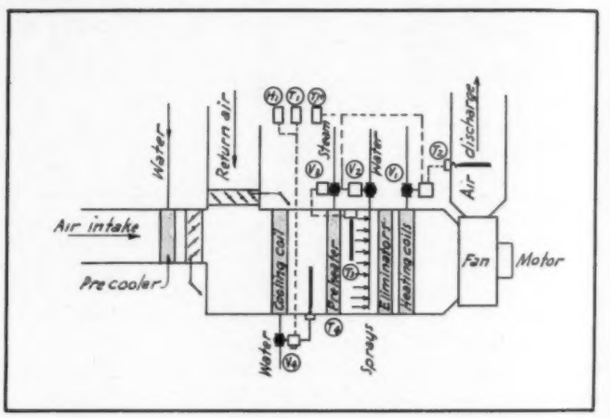


Cycle—The precooling is usually accomplished by city or well water at a temperature above 60 F. No control is usually included on this coil, as the lowest temperature possible to obtain from this coil would be higher than the leaving condition from the sensible and latent cooling coils.

The controls for the heating, ventilating, and humidifying cycle and for the cooling, dehumidifying, and ventilating cycle are exactly the same as shown in the second example.

As an alternate on this system, a face and bypass damper on the reheating coil might be included, thus giving more accurate control for heating. Thermostat T_4 would then control the damper motor operating the face and bypass damper. The damper motor controlling the face and bypass damper would also be equipped with an auxiliary contact to close valve V_5 in the steam supply line to the reheating coil. V_5 would then be of the two position type instead of the modulating type, as described in the second example.

The following system incorporates several differences from those previously discussed:



Cycle—For heating, a manually operated damper in the outside air intake allows adjustment of the amount of outside air taken into the system, and a manually operated damper in the return air duct work permits adjusting the amount of return air recirculated by the fan. Insertion type thermostat T_3 after the preheating coil controls the amount of steam supplied to the preheating coil by means of modulating type motorized or diaphragm valve V_3 in the steam supply line to this coil, thus maintaining a constant leaving air temperature from this preheater.

Motorized or diaphragm valve V_2 in the water supply line to the sprays is controlled by means of combination temperature and humidity controller TH_1 which also controls motorized or diaphragm valve V_1 (which is of the modulating type) in the steam supply line to the reheating coil.

V_1 then supplies the correct amount of heat to the air after it has been humidified to maintain constant room temperature. V_1 is also interconnected with insertion type thermostat T_2 which is set to maintain a constant air temperature from the fan, regardless of the setting of TH_1. That is, thermostat T_2 acts as a low limit control, preventing the discharge of cold air.

The controls for cooling consist of the same manually operated dampers in the outside air intake and return air duct for control of the outside and recirculated air. Room thermostat T_1 controls the operation of the motorized or diaphragm valve V_4 in the refrigerant return line from the cooling coil. V_4 is of the modulating type. T_4 also controls the operation of V_4 so that a set air temperature leaving the cooling coil is maintained.

Room humidistat H_1 is also interconnected with V_4 so that in case the room is down to temperature, but the humidity is too high, room humidistat H_1 assumes control of V_4.

As an alternate, this room thermostat and room humidistat might be replaced by a combination temperature and humidity controller similar to TH_1.

During the cooling cycle the preheating coil, the reheating coil, and the sprays are all shut down manually.

Four More Tables of Pipe Properties Comprise This Month's Heating ✦ Piping and Air Conditioning Summary Sheets

[Pages 1-4 of these tables were published last month]

Properties of Schedule 80 Pipe

Nominal Pipe Size In.	Outside Diam In. D	Wall Thick-ness In. t	Inside Diam In. d	Inside Diam Square In. d^2	Inside Diam Power d^3	Area of Metal Sq.In.	Cross-Sectional Area Inside Diam Sq. In.	Cross-Sectional Area Outside Diam Sq. Ft.	External Surface Per Ft Sq. Ft.	Moment of Inertia In.^{4}	Weight of Pipe Per Ft Lb	Weight of Water Per Ft of Pipe Lb
1/8	0.405	0.095	0.215	0.0462	0.00993	0.063	0.0363	0.000629	0.106	0.000218	0.32	0.0159
1/4	0.540	0.119	0.302	0.0913	0.00276	0.157	0.0716	0.00126	0.141	0.003766	0.54	0.031
3/8	0.675	0.126	0.423	0.1789	0.01324	0.217	0.1406	0.00098	0.177	0.006819	0.74	0.0609
1/2	0.840	0.147	0.546	0.2981	0.04668	0.363	0.2341	0.00163	0.220	0.02808	1.09	0.1013
3/4	1.050	0.154	0.742	0.5506	0.22849	0.433	0.4324	0.00300	0.275	0.06479	1.49	0.1876
1	1.315	0.179	0.957	0.9158	0.80279	0.639	0.7193	0.00499	0.344	0.1056	2.17	0.3118
1 1/4	1.660	0.191	1.278	1.633	3.409	0.861	1.283	0.00891	0.435	0.2618	3.00	0.5555
1 1/2	1.900	0.200	1.500	2.250	7.594	1.068	1.767	0.01225	0.498	0.3918	3.64	0.7648
2	2.375	0.218	1.939	3.760	27.41	1.477	2.953	0.02050	0.622	0.8679	5.03	1.279
2 1/2	2.875	0.276	2.323	5.396	67.44	2.254	4.238	0.02942	0.753	1.924	7.67	1.834
3	3.500	0.300	2.900	8.410	205.1	3.016	6.605	0.04589	0.917	3.894	10.3	2.899
3 1/2	4.0	0.318	3.364	11.32	430.8	3.678	8.891	0.06170	1.047	6.280	12.5	3.847
4	4.5	0.337	3.826	14.64	819.8	4.407	11.50	0.07986	1.178	9.610	15.0	4.976
5	5.563	0.375	4.813	23.16	2,583	6.112	18.19	0.1263	1.456	20.69	20.8	7.875
6	6.625	0.432	5.761	33.18	6,346	8.405	26.07	0.1810	1.734	40.49	28.6	11.29
8	8.625	0.500	7.625	58.14	25,775	12.76	45.66	0.3171	2.258	105.7	43.4	19.79
10	10.75	0.593	9.564	91.49	80,080	18.92	71.84	0.4988	2.815	244.8	64.4	31.13
12	12.75	0.687	11.376	129.41	190,863	26.03	101.64	0.7056	3.338	475.1	88.6	44.04
14 OD	14.0	0.750	12.500	156.25	305,176	31.23	122.72	0.8522	3.665	607.3	107	53.18
16 OD	16.0	0.843	14.314	204.89	600,804	40.14	160.98	1.1178	4.189	1,256	137	69.73
18 OD	18.0	0.937	16.126	260.05	1,090,918	50.23	204.24	1.4185	4.712	1,633	171	86.50
20 OD	20.0	1.031	17.938	321.79	1,857,540	61.44	252.72	1.7550	5.236	2,772	209	109.51
24 OD	24.0	1.218	21.564	465.01	4,648,799	87.17	365.22	2.5362	6.283	5,672	297	158.26

Properties of Schedule 160 Pipe

Nominal Pipe Size In.	Outside Diam In. D	Wall Thick-ness In. t	Inside Diam In. d	Inside Diam Square In. d^2	Inside Diam Power d^3	Area of Metal Sq.In.	Cross-Sectional Area Inside Diam Sq. In.	Cross-Sectional Area Outside Diam Sq. Ft.	External Surface Per Ft Sq. Ft.	Moment of Inertia In.^{4}	Weight of Pipe Per Ft Lb	Weight of Water Per Ft of Pipe Lb
1/2	0.840	0.187	0.466	0.2172	0.01297	0.3636	0.1706	0.00118	0.220	0.02218	1.31	0.094
3/4	1.060	0.218	0.624	0.3970	0.06726	0.5698	0.2961	0.00206	0.275	0.05289	1.94	0.170
1	1.315	0.250	0.815	0.6642	0.3598	0.8365	0.5217	0.00362	0.344	0.1251	2.85	0.230
1 1/4	1.660	0.250	1.160	1.346	2.100	1.107	1.069	0.00734	0.435	0.2859	3.77	0.46
1 1/2	1.900	0.281	1.338	1.790	4.288	1.429	1.406	0.00976	0.498	0.4324	4.86	0.61
2	2.375	0.343	1.689	2.853	13.74	2.190	2.241	0.01566	0.622	1.162	7.45	0.97
2 1/2	2.875	0.375	2.125	4.516	43.33	2.945	3.546	0.02463	0.753	2.353	10.0	1.54
3	3.5	0.437	2.626	6.896	124.9	4.205	5.416	0.03761	0.917	5.026	14.3	2.25
3 1/2	4.0											
4	4.5	0.531	3.438	11.82	400.3	6.621	9.283	0.06447	1.178	13.27	22.6	4.02
5	5.563	0.625	4.313	18.60	1,492	9.696	14.61	0.10153	1.456	30.03	33.0	6.33
6	6.625	0.718	5.189	26.93	3,782	13.28	21.13	0.1469	1.734	58.97	45.3	9.16
8	8.625	0.906	6.813	46.42	16,079	21.97	36.46	0.2532	2.257	165.9	74.7	15.80
10	10.75	1.125	8.500	72.25	44,371	34.02	56.75	0.3941	2.815	399.3	116	24.59
12	12.75	1.312	10.126	102.54	106,661	47.14	80.53	0.5592	3.338	781.1	161	34.89
14 OD	14.0	1.406	11.188	125.17	175,992	55.63	98.31	0.6827	3.665	1,117	190	42.60
16 OD	16.0	1.562	12.876	165.79	353,980	70.95	130.21	0.9043	4.189	1,868	241	56.43
18 OD	18.0	1.750	14.500	210.25	660,973	89.84	165.13	1.1467	4.712	2,983	304	71.55
20 OD	20.0	1.938	16.124	260.05	1,090,919	109.9	204.84	1.4183	5.236	4,534	374	88.50
24 OD	24.0	2.312	19.376	375.43	2,750,993	157.5	294.96	2.0476	6.283	9,349	536	127.77

Here is the third installment of the new series of "summary sheets," on this and the preceding page. The purpose of these sheets is to make available for quick reference for everyday use information needed in heating, piping and air conditioning work. . . . For complete information, refer to the original article from which these data were obtained, as the purpose of these summaries is not to take the place of the article, but rather to put the essentials into convenient reference form. . . . The Editors will appreciate your suggesting what type of material you'd like presented in this way. . . . Address HEATING, PIPING AND AIR CONDITIONING, 6 North Michigan Avenue, Chicago, Ill.

Properties of Schedules 60 and 100 Pipe

Nominal Pipe Size in.	Outside Diam in. D	Wall Thickness in. t	Inside Diam in. d	Inside Diam Squared d^2	Inside Diam Fifth Power d^5	Area of Metal Sq.In.	Internal Cross-Sectional Area Sq. In.	Sq. Ft.	External Surface Per ft Sq. Ft.	Moment of Inertia $in.^4$	Weight of Pipe Per ft Lb	Weight of Water Per ft of Pipe Lb
					SCHEDULE 60							
8	8.625	0.406	7.813	61.04	29,113	10.48	47.94	0.3329	2.258	88.73	35.7	20.77
10	10.75	0.500	9.75	95.06	88,110	16.10	74.66	0.5185	2.814	212.0	54.8	32.35
12	12.75	0.561	11.626	135.16	212,399	21.52	106.16	0.7372	3.338	400.4	73.2	46.00
14 OD	14.0	0.593	12.814	164.20	345,480	24.98	138.96	0.8956	3.665	562.3	85.0	55.86
16 OD	16.0	0.656	14.688	215.74	663,618	31.62	169.44	1.1766	4.189	932.4	108	73.42
18 OD	18.0	0.718	16.564	274.37	1,246,884	38.98	215.49	1.4964	4.712	1,458	133	93.38
20 OD	20.0	0.812	18.376	337.68	2,065,342	48.95	265.21	1.8417	5.236	2,257	167	114.92
24 OD	24.0	0.937	22.126	489.56	5,302,913	67.89	384.50	2.6701	6.283	4,521	231	166.61
					SCHEDULE 100							
8	8.625	0.593	7.439	55.34	22,781	14.96	43.46	0.3018	2.258	121.3	50.9	18.83
10	10.75	0.718	9.314	86.75	69,357	22.63	68.13	0.4732	2.814	286.1	77.0	29.53
12	12.75	0.843	11.064	122.41	165,791	31.53	96.14	0.6677	3.338	561.6	108	41.64
14 OD	14.0	0.937	12.126	147.04	362,173	38.45	115.49	0.8020	3.665	824.4	131	50.04
16 OD	16.0	1.031	13.938	194.27	826,080	48.48	153.56	1.0596	4.189	1,364	165	66.13
18 OD	18.0	1.156	15.688	246.11	950,250	61.17	193.30	1.3433	4.712	2,160	208	83.76
20 OD	20.0	1.250	17.50	306.25	1,641,309	73.63	240.53	1.6703	5.236	3,250	251	104.23
24 OD	24.0	1.500	21.00	441.00	4,084,101	106.03	346.36	2.4055	6.283	6,740	361	150.09

Properties of Schedules 120 and 140 Pipe

Nominal Pipe Size in.	Outside Diam in. D	Wall Thickness in. t	Inside Diam in. d	Inside Diam Squared d^2	Inside Diam Fifth Power d^5	Area of Metal Sq.In.	Internal Cross-Sectional Area Sq. In.	Sq. Ft.	External Surface Per ft Sq. Ft.	Moment of Inertia $in.^4$	Weight of Pipe Per ft Lb	Weight of Water Per ft of Pipe Lb
					SCHEDULE 120							
4	4.5	0.437	3.626	13.15	626.8	5.578	10.33	0.0717	1.178	11.65	19.0	4.47
5	5.563	0.500	4.563	20.82	1,978	7.953	16.35	0.1136	1.456	25.73	27.1	7.09
6	6.625	0.562	5.501	30.26	5,037	10.70	23.77	0.1650	1.734	49.61	36.4	10.30
8	8.625	0.718	7.189	51.68	19,202	17.84	40.59	0.2819	2.257	140.5	60.7	17.59
10	10.75	0.843	9.064	82.16	61,179	28.24	64.53	0.4481	2.817	324.2	89.2	27.96
12	12.75	1.000	10.750	115.56	143,563	36.91	90.76	0.6303	3.338	641.6	126	39.33
14 OD	14.0	1.062	11.876	141.04	236,239	43.17	110.77	0.7693	3.665	909.3	147	48.00
16 OD	16.0	1.218	13.564	183.98	459,133	56.56	144.50	1.0035	4.189	1,555	193	62.62
18 OD	18.0	1.343	15.314	234.52	842,254	70.88	184.19	1.2791	4.712	2,453	239	79.81
20 OD	20.0	1.500	17.000	289.00	1,419,857	67.18	226.98	1.5762	5.236	3,754	297	98.35
24 OD	24.0	1.750	20.500	420.25	3,680,506	122.33	330.06	2.2921	6.283	7,617	416	143.03
					SCHEDULE 140							
8	8.625	0.812	7.001	49.01	16,819	19.93	38.50	0.2673	2.257	153.7	67.8	16.68
10	10.75	1.000	8.750	76.56	51,291	30.63	60.13	0.4176	2.817	367.8	105	26.06
12	12.75	1.125	10.500	110.25	127,628	41.08	86.59	0.6013	3.338	700.5	140	37.52
14 OD	14.0	1.250	11.500	132.25	201,136	50.07	103.87	0.7213	3.665	1,027	171	45.01
16 OD	16.0	1.437	13.126	172.29	389,670	65.74	135.32	0.9397	4.189	1,760	224	58.64
18 OD	18.0	1.562	14.876	221.30	728,502	80.66	173.80	1.2070	4.712	2,749	275	75.32
20 OD	20.0	1.750	16.500	272.25	1,222,981	100.33	213.82	1.4849	5.236	4,216	342	92.66
24 OD	24.0	2.062	19.876	395.06	3,108,022	142.11	310.28	2.1547	6.283	8,625	484	134.45

Heating·Piping
and Air Conditioning

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 F and 46 F

(Part 1)

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THE investigation in summer cooling in the Research Residence during the summer of 1935[1] made use of water at a temperature of from 58 F to 60 F. The investigation for the summer of 1936 was undertaken to determine to what extent water at temperatures of approximately 52 F and 46 F could be used to produce satisfactory cooling and dehumidification in the Research Residence when supplemented by the circulation of outdoor air through the second story at night and when approximately one air change per hour of outdoor air was used for the purpose of ventilation during the day.

Description of the Research Residence and Cooling Equipment

The Research Residence together with the forced-air heating system has been described in a previous paper.[2] For the purpose of this investigation the Residence was equipped with awnings at all east, south, and west windows, and the sun-room was isolated from the rest of the house by closing the doors leading into the dining room. The entire third story was regarded as an attic and during the daytime was isolated from the rest of the house by closing the door at the head of the stairs. The attic windows, however, were opened to provide ventilation in the attic both day and night. The total space cooled amounted to 14,170 cu ft. Unless otherwise specified, the state of the Residence was the same for the work done during the two summers.

The arrangement[3] of the forced-air duct system and fan was also the same for the two summers, and in each case, all return ducts with the exception of the central one containing the cooling coil were blocked. The delivery ducts to the sun-room and third story were blocked, and the dampers in the ducts to the first and second stories were adjusted to maintain the proper balance between the amounts of air required for cooling on these two stories.

The arrangement of the cooling plant is shown in Fig. 1. The cooling coil consisted of 8 rows of finned tubes and one row of tubes without fins. Each row consisted of 12 tubes connected into headers or return bends at the ends of the tubes. These headers were connected in such a way that the water flowed across the duct through one row of 12 tubes and returned through the next row, thus alternating through each succeeding row of tubes. The water connections were made so that the flow of the water was counter to the flow of the air, thus providing that the coldest air came in contact with the tubes containing the coldest water. The coil was arranged so that the air flowed horizontally. The water passed through a calibrated water meter before entering the coil. The tubes were $\frac{5}{8}$ in. outside diameter and had 8 fins per inch. The overall diameter including the fins was $1\frac{7}{16}$ in. As arranged in the duct, the gross face section of the coil was $17\frac{9}{16}$ in. high and $35\frac{1}{2}$ in. wide, giving a gross face area of 4.33 sq ft. The net face area was 3.56 sq ft. The net free area was 1.76 sq ft. The total area of cooling surface, including the row without fins was 429 sq ft, and, based on a volume of air circulated of 1231 cfm, the gross face velocity was 284, the net face velocity was 346, and the velocity through the net free area was 700 fpm.

For the cooling season of 1935, during which the tests were conducted with water supplied to the coil at a temperature of 58 F, the selection of the cooling coil was based on the following considerations: In selecting the coil required to absorb the maximum cooling load on the Research Residence, it was found that there was considerable latitude in the choice, provided that no restrictions were placed on the amount of cooling water, or the amount of coil surface used. The cooling load was estimated as 20,000 Btu per hour sensible heat and 7,000 Btu per hour latent heat. With 80 F maintained in the rooms, the restriction of 70 F for the temperature of the air leaving the registers, together with the known rise of 4 F in the duct system, determined that the temperature drop of the air passing through the coil should be 15 F; or from 81 F to 66 F. This drop

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[1] A.S.H.V.E. Research Paper, Study of Summer Cooling in the Research Residence Using Water from the City Water Mains, A. P. Kratz, M. K. Fahnestock, S. Konzo, and E. L. Broderick, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping, and Air Conditioning*, May, 1936, pp. 278-288.

[2] A.S.H.V.E. Research Paper, Study of Summer Cooling in the Research Residence at the University of Illinois, A. P. Kratz and S. Konzo, A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 95.

[3] Loc. cit. Note 1.

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in temperature, together with the sensible heat load, limited the air requirement to 1300 cfm. A net face velocity of 365 fpm permitted the use of an air duct of reasonable dimensions, and thus fixed the face area of the coil. The freedom of choice then proved to be in the amount of coil surface, or the number of rows of coil to be used.

The data furnished by the manufacturer of the coil indicated that the required temperature drop in the air could be obtained either by the use of a large amount of cooling water with a small amount of coil surface, or by a small amount of water used in connection with a large amount of coil surface. Furthermore, the resistance to the flow of air increased rapidly as the amount of surface, or number of rows of coil, increased. In addition to the possible limit imposed on the air delivery by the higher resistance, the use of a large amount of surface has the further disadvantage of high first cost. On the other hand, the use of a small amount of coil surface has the disadvantage of high operating cost due to the amount of water used. Under these conditions a compromise is necessary. In the case of the Research Residence, this was effected by selecting 8 rows of coil for the tests with 58 F water. This determined a water rate such that a decrease of one or two rows would have resulted in an appreciable increase in water consumption, while an increase of one or two rows would not have resulted in an appreciable decrease in water consumption.

The resistance offered by the coil itself to the flow of water was only 0.5 lb per square inch. The drop in pressure from the city main to the entrance of the coil, however, was approximately 25 lb per square inch through the ¾ in. line from the main to the coil. This loss in pressure may in some cases be the limiting factor in determining the amount of water available for cooling purposes. The estimated resistance to flow of air of the 8-row coil was 0.29 in. of water, and this was regarded as the upper limit for the system under consideration.

For the cooling season of 1936, during which the tests were conducted with water supplied to the coil at temperatures of approximately 52 F and 46 F, a similar compromise was effected. The use of the 8 rows of cooling coil previously employed would make possible the use of a small quantity of water while the use of a lesser number of rows of coil would require a large quantity of water. For the latter case the initial cost of the equipment would be lower than for the former case, but the operating cost would be higher as a result of the less economical use of the water. Since a small reduction in the initial cost of the cooling equipment would be offset over a period of years by the reduced annual operating cost, it was considered advisable to retain the same number of coil sections used in the tests with 58 F water and to make suitable reductions in the quantity of water used. Hence, the tests for the season of 1936 were made with rates of water flow of approximately 260 gal per hour and 165 gal per hour with water temperatures of 52 F and 46 F, respectively, as compared with the 360 gal per hour used in connection with the 58 F water.

The operation of the cooling plant was controlled by means of a thermostat located at the 60-in. level in the hall on the second story. As shown in Fig. 2 this thermostat was used in connection with a motor-operated, water circulator which was placed on the inlet side of the cooling coil and which was used to start and stop the flow of water through the coil. The temperature of the water entering the coil was maintained at a constant value by means of the three-way mixing valve located on the inlet side of the circulating pump. This valve served to proportion and mix the warm water received from the outlet of the cooling coil and the cooled water received from the heat exchangers so that the temperature of the water entering the cooling coil was maintained at a predetermined value of either 52 F or 46 F. The operation of the three-way valve was automatic, and was controlled by means of a regulating thermostat placed in the

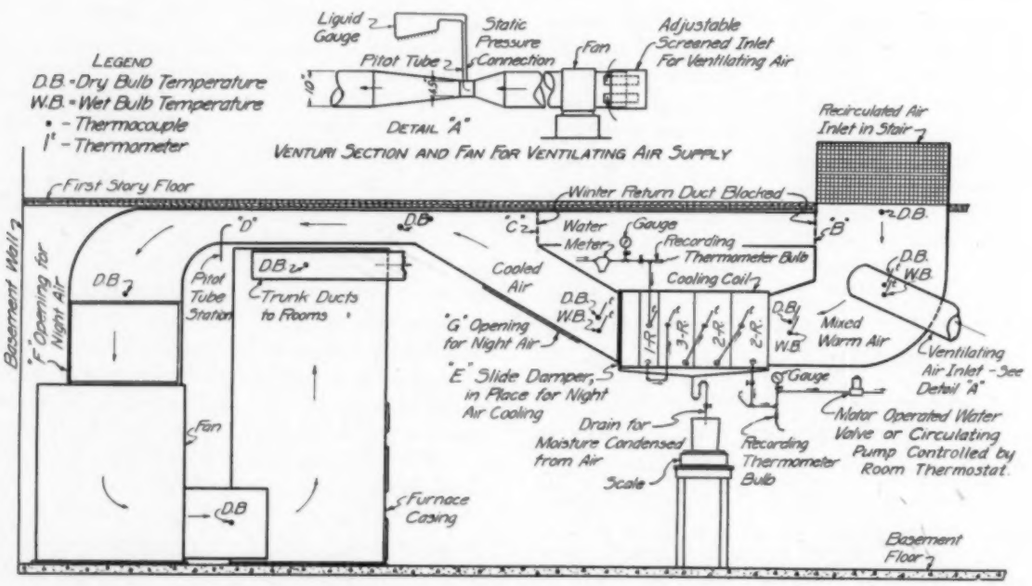

Fig. 1—Diagram of cooling plant with water cooling coil

water circuit on the outlet side of the water circulator, as indicated in Fig. 2.

For the tests conducted in 1936, since water from the city water mains was not available at temperatures of less than 58 F, it was necessary to provide a reservoir of water that was cooled to a predetermined temperature slightly lower than the temperature of water entering the cooling coil. The auxiliary equipment which was required to maintain the proper temperature in the reservoir is shown in the right hand portion of Fig. 2. It may be observed that the warm water which left the cooling coil was passed through heat exchangers by means of a circulating pump, was cooled, and was then returned to the reservoir from which the cold water supply to the cooling coil was obtained. The heat exchangers consisted of two shell-and-tube type condensing units in which the warm water passed through the tubes and the refrigerant was circulated over the outside of the tubes. The auxiliary equipment provided a very satisfactory means of obtaining the desired quantity and temperature of water for the cooling coil and was an excellent substitute for a natural cold water supply.

In order to provide for both heating in the winter and cooling in the summer the cooling coil was installed in a by-pass in the central return duct as shown in Fig. 1. For the summer work, the duct was blocked with tightly fitting dampers at *B* and *C*, and all of the air delivered by the fan in the forced-air heating system passed through the cooling coil when the air in the house was being recirculated. For the purpose of providing outdoor air for cooling during the night, a slide damper, *E*, was placed in the by-pass on the outlet side of the coil, and a door, *G*, was placed in the recirculating duct on the down-stream side of the slide damper. When outdoor air was required, the basement door and the door, *G*, in the recirculating duct were opened, and the slide damper, *E*, was closed. The fan in the forced-air system delivered approximately 1231 cfm of air when recirculating the air in the house and 2220 cfm when using outdoor air at night. These air volumes were based on an air density of 0.0749-lb per cubic foot. Outdoor air, for the purpose of ventilation during the day, was provided by means of the duct shown as Detail *A* in Fig. 1.

The study of baseboard and sidewall registers was made in the east bedroom of the Research Residence. This room was 13½ ft by 20 ft, with an 8 ft-4 in. ceiling, and had three exposed walls; north, south, and east. It had one door, which remained open during all of the tests.

The location and arrangement of the warm-air stack and registers are shown in Figs. 3 and 11. The sidewall register was the same size as the baseboard register and was placed at the head of the extension of the stack for the baseboard register in order to facilitate changes from one position to the other. The same register face was used in both locations, and consisted of horizontal louvers, 1 in. in width, which could be set at any angle from 0 deg to 90 deg with the horizontal. In the latter case the louvers were completely closed. When the sidewall register was in use, the baseboard outlet was sealed with a flat plate placed with the inner face flush with the wall of the stack. Similarly, when the baseboard register was in use, the stack was sealed just above the baseboard outlet. Hence, in either case, the stack was continuous and without offsets or projections.

Method of Conducting Tests

Continuous records were made during the summer of air temperatures in the various parts of the house and in the different sections of the duct system. Other incidental temperatures and relative humidities were also read at regular intervals. A complete description of the measurements which were made has been given in the previous report.[4]

During the periods of operation, observations were made of the weight of water circulated through the cooling coil, the temperature of the water entering and leaving the coil, the weight of water condensed from the air, and the electrical input to the fan motor. The weight of water circulated was obtained by means of a calibrated water meter and the air quantities were obtained from traverses made with Pitot tubes at section *D* in the central return duct and at the venturi section in the ventilating air duct as shown in Fig. 1. Incidental data from the operation of the heat exchangers were also obtained.

For all of the tests the Residence was operated on the following schedule:

[4]Loc. cit. Note 1.

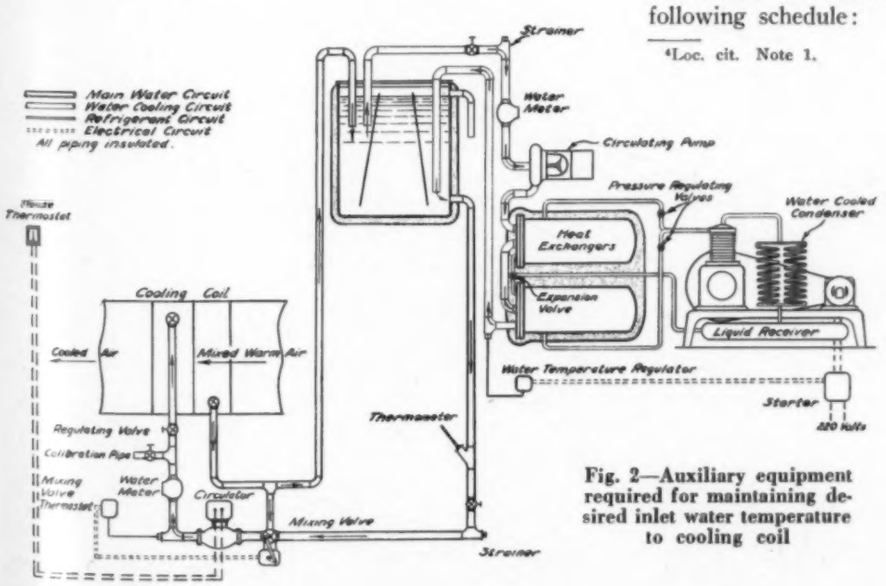

Fig. 2—Auxiliary equipment required for maintaining desired inlet water temperature to cooling coil

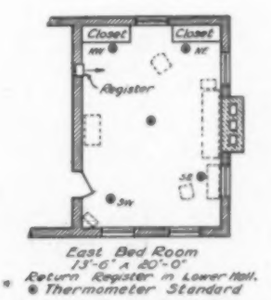

Fig. 3—Location of register with respect to exposed walls

Table 1—Typical Operating Data and Results Obtained by Cooling with Water at 2:00 P. M. on July 7 and July 13, 1936

1.	Date	7-7-36	7-13-36	13.	Air temp. rise in ducts and casing, F...	3.7	3.8
2.	Test No.	12	18	14.	Basement air temp. at br. level, F.....	74.0	74.2
3.	Series	1-36	2-36	15.	Quantity of air through wet coil		
4.	Outdoor Air				Cfm	1231	1231
	Dry Bulb, F	103.8	103.8		Pounds	5540	5540
	Wet Bulb, F	75.8	76.8		Density of air, pounds per cubic foot.	0.075	0.075
	Dew Point, F	63.8	65.7	16.	No. of house air recirculations per hour	5.2	5.2
	Relative Humidity, per cent	27.8	29.5	17.	Ventilating air		
	Specific Humidity, grains per pound dry air	88.4	96.2		Cfm	226	226
5.	Indoor Air, average br. level temp.				Pounds	1016	1016
	1st Story, F	79.4	79.5		Density of air, pounds per cubic foot.	0.075	0.075
	2nd Story, F	81.6	80.8	18.	Cooling Coil		
6.	Indoor Air, house average at br. level				Gross Face Area, sq ft	4.33	4.33
	Dry Bulb, F	80.3	80.1		Net Free Area, sq ft	1.76	1.76
	Wet Bulb, F	67.6	67.4		Surface of Coil, sq ft	429	429
	Dew Point, F	61.0	61.0	19.	Air Velocity through Coil		
	Relative Humidity, per cent	51.8	52.0		Gross Face, fpm	284	284
	Specific Humidity, grains per pound dry air	80.8	80.5		Net Face, fpm	346	346
7.	Indoor Air, entering return grille				Free Area, fpm	700	700
	Dry Bulb, F	78.9	78.8	20.	Moisture condensed from air, pounds per hour	2.26	2.24
	Wet Bulb, F	67.3	66.8	21.	Heat given up by air; total Btu per hour	24970	25630
	Dew Point, F	61.5	60.5		Heat due to moisture in air		
	Relative Humidity, per cent	55.0	53.3		Btu per hour	2370	2350
	Specific Humidity, grains per pound dry air	82.0	78.5		Per cent of total heat	9.5	9.2
8.	Ventilating Air				Sensible heat		
	Dry Bulb, F	97.8	99.0		Btu, per hour	22600	23280
	Wet Bulb, F	72.8	71.4		Per cent of total heat	90.5	90.8
	Dew Point, F	61.0	57.6	22.	Water temp. in cooling coil		
	Relative Humidity, per cent	30.0	25.6		Inlet, F	51.9	46.9
	Specific Humidity, grains per pound dry air	80.7	71.0		Outlet, F	63.4	65.4
9.	Mixed Air, entering cooling coil			23.	Rise in water temp. through coil		
	Dry Bulb, F	81.0	81.4		First row, F	1.1	2.6
	Wet Bulb, F	67.1	66.6		First four rows, F	5.3	9.7
	Dew Point, F	59.8	58.7		First six rows, F	8.2	13.8
	Relative Humidity, per cent	47.7	45.2		Total coil, eight rows, F	11.5	18.5
	Specific Humidity, grains per pound dry air	77.9	73.8	24.	Water pressure at coil, pounds per sq in	4.5	4.5
10.	Mixed Air, leaving cooling coil			25.	Quantity of cooling water		
	Dry Bulb, F	64.0	63.9		Pounds per hour	2100	1344
	Wet Bulb, F	60.8	60.0		Gallons per hour	252	161
	Dew Point, F	58.9	57.7	26.	Heat absorbed by water, Btu per hour	24150	24860
	Relative Humidity, per cent	83.3	79.9	27.	Recirculating fan speed, rpm	445	445
	Specific Humidity, grains per pound dry air	74.1	71.0	28.	Fan motor data		
11.	Air temp. drop through cooling coil, F..	17.0	17.5		Horse power	¾	¾
12.	Temp. cooled air leaving registers, average, F	67.7	67.7		Power rate, watts	440	440
				29.	Static pressure loss through coil		
					Coil, wet, in. water	0.28	0.28
					Coil, dry, in. water	0.23	0.23
				30.	Total resistance of system, in. water...	0.51	0.51

The second story windows and the attic and basement doors were closed at 7 a.m. and the dampers were changed so that the fan, which had been delivering outdoor air through the system, started delivering recirculated air and the outdoor air admitted for ventilation. The former was equivalent to 4.2 recirculations of the air in the house and the latter was equivalent to one air change per hour, making a total of 5.2 air changes per hour delivered by the fan. The fan was allowed to run continuously during the day. When the temperature of the indoor air on the second story rose to 81 F the motor driven water circulator, actuated by the thermostat, admitted water to the cooling coil. The cooling plant was allowed to operate with thermostatic *on* and *off* control maintaining 81 F on the second story until the effective temperature outdoors became equal to the effective temperature on the second story indoors. The water circulator was then stopped; eleven of the windows on the second story and the attic door were opened; and the dampers were set and the basement door opened, so that the fan delivered outdoor air through the duct system, the fan continuing to run until 7 a.m. The fan delivery was 2220 cfm or 9.4 air changes per hour.

In the first series of tests, which has been designated as Series 1-36, the temperature of the water which was supplied to the cooling coil was maintained at approximately 52 F and the quantity of water circulated through the coil was regulated so that the flow was approximately 260 gal per hour. In the second series of tests, designated as Series 2-36, the water temperature was

maintained at approximately 46 F and the quantity of the water flowing was adjusted to a value of approximately 165 gal per hour.

During all tests made in the study of baseboard and sidewall registers, the cooling plant was operated normally to maintain approximately 80 F at the 5 ft level in the rooms of the Residence. Observations were then made of the velocity and temperature of the air leaving the register, and of the air temperature on the central axis of the room 4 in. above the floor, 5 ft above the floor, and 4 in. below the ceiling. In addition, for some of the tests the air temperatures at two levels, 4 in. above the floor and 5 ft above the floor, were observed in all the four corners of the room, as shown in Fig. 3. For each register arrangement a large number of observations was made to include a wide range of outdoor temperatures. The temperatures at the different levels were plotted against the difference between the outdoor temperature and the temperature at the 5 ft level in the room.

Results of Tests

General Conditions

The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the results obtained on a typical day. For this purpose the test made on July 7, 1936, was selected as typical of the operation obtained with water at a temperature of 52 F (Series 1-36) and the results are shown in the first column of Table 1 and in Fig. 4. The test made on July 13, 1936, was selected as typical of the operation obtained with water at a temperature of 46 F (Series 2-36) and the results are shown in the second column of Table 1. General results from all the tests are given in Table 2.

On the day which was selected as being typical for the tests made under conditions of operation for Series 1-36, the outdoor temperature was 103.8 F at 2 p.m. and reached a maximum of 104.0 F at 3 p.m. At this time

Table 2—Complete Data and Results for Tests

Date 1936	Test No	Series No.	Weather Data: Degree-Hours Above 85°	Weather Data: Degree-Hours Above 90°	Weather Data: Sun Hours: Minutes	Outdoor Air: Maximum Temp. During Test F	Outdoor Air: Minimum Temp. Previous to Test F	Outdoor Air: Average During Period Requiring Cooling with Water: Temp. F	Outdoor Air: Average During Period Requiring Cooling with Water: Hum. %	Indoor Air Conditions: At End of Night Cooling: Temp. F	Indoor Air Conditions: At End of Night Cooling: Hum. %	Indoor Air Conditions: Average During Period Requiring Cooling with Water: Temp. F	Indoor Air Conditions: Average During Period Requiring Cooling with Water: Hum. %
1	2	3	4	5	6	7	8	9	10	11	12	13	14
6-17-36	1	1-36	66.5	25.4	11:05	95.0	67.0	88.2	51.7	74.2	52.0	78.8	48.1
6-20-36	2	1-36	47.0	10.2	15:07	94.0	68.0	84.8	49.8	73.7	55.1	79.1	49.9
6-27-36	3	1-36	19.1	0.0	5:10	89.0	69.0	83.8	54.8	76.0	46.9	77.9	53.5
6-28-36	4	1-36	43.8	6.4	13:20	93.0	68.0	88.8	36.8	76.7	62.6	79.1	49.3
6-29-36	5	1-36	137.0	68.9	8:15	101.0	60.0	87.4	44.1	70.8	40.8	78.7	52.4
7- 2-36	7	1-36	28.9	0.0	10:46	92.0	67.0	85.0	30.4	73.8	63.3	79.2	48.5
7- 3-36	8	1-36	34.8	0.3	15:06	91.0	64.0	86.7	37.1	73.7	66.5	79.0	53.0
7- 4-36	9	1-36	125.0	63.3	15:02	100.0	72.0	88.4	41.3	76.9	64.0	79.0	54.5
7- 5-36	10	1-36	142.1	75.5	13:47	102.0	73.0	90.8	40.2	.....	.....	78.8	55.4
7- 6-36	11	1-36	174.3	103.0	14:53	105.0	79.0	91.6	38.6	.....	.....	79.5	54.6
7- 7-36	12	1-36	194.5	117.8	14:25	104.0	78.0	93.2	36.7	.....	.....	79.8	52.6
7- 8-36	13	1a-36	170.2	107.2	14:59	104.0	80.0	97.2	32.8	.....	.....	80.3	51.2
7- 9-36	14	1a-36	93.9	44.0	11:34	99.0	72.0	87.8	43.1	79.0	61.8	78.9	56.5
7-10-36	15	2-36	163.6	97.3	10:34	104.0	76.0	91.3	39.9	.....	.....	79.2	53.3
7-11-36	16	2-36	184.3	111.7	11:48	105.0	79.0	92.8	37.4	.....	.....	79.5	52.9
7-12-36	17	2-36	201.6	125.5	13:52	106.0	79.0	92.0	39.1	.....	.....	79.7	52.3
7-13-36	18	2-36	180.9	101.2	12:47	105.0	75.0	92.4	38.9	.....	.....	79.7	54.2
7-14-36	19	2-36	207.6	135.4	14:88	108.0	79.0	93.0	35.6	.....	.....	79.7	50.5
7-15-36	20	2-36	93.0	57.4	5:05	107.0	77.0	86.6	49.5	.....	.....	78.8	57.9
7-16-36	21	2-36	83.1	31.4	14:50	98.0	74.0	90.2	40.1	.....	.....	78.9	56.1
7-17-36	22	2-36	108.9	52.6	14:48	100.0	68.0	91.6	31.3	75.9	58.4	78.8	50.2
7-18-36	23	2-36	0.9	0.0	6:00	92.0	75.0	83.5	44.1	77.4	76.9	79.1	58.7
7-23-36	24	2-36	98.6	47.5	11:51	97.0	73.0	92.4	36.9	76.8	71.1	79.4	56.8
7-24-36	25	2-36	8.0	0.0	11:18	92.0	67.0	83.3	41.4	75.5	63.7	79.2	56.3
7-25-36	26	2-36	18.1	0.0	9:00	90.0	66.0	84.6	62.2	74.1	76.6	79.3	71.9
7-26-36	27	2-36	98.3	48.0	12:23	99.0	76.0	88.0	47.0	79.0	72.1	79.3	60.6
7-27-36	28	2-36	107.0	60.5	12:10	101.0	77.0	93.6	38.3	.....	.....	79.7	54.4
7-28-36	29	2-36	33.2	4.7	11:22	93.0	70.0	87.8	43.1	77.1	72.9	79.5	59.3
8- 3-36	30	2-36	12.1	0.0	10:26	87.0	68.0	82.9	48.1	74.8	66.5	79.8	59.5
8- 9-36	31	2-36	50.4	10.5	14:02	93.0	68.0	86.1	46.3	71.5	58.8	79.3	59.8
8-10-36	32	2-36	5.6	0.0	11:41	87.0	67.0	83.9	46.2	75.3	73.0	79.5	64.4
8-13-36	33	2-36	88.3	35.2	13:30	97.0	71.0	88.9	45.1	75.8	69.1	79.1	59.1
8-14-36	34	2-36	18.8	0.6	6:06	90.0	71.0	82.7	50.1	76.9	72.8	79.6	63.9
8-15-36	35	2-36	68.7	23.0	13:51	95.0	77.0	84.3	53.6	.....	.....	78.9	64.1
8-16-36	36	2-36	98.4	16.2	13:48	95.0	72.0	88.2	49.4	.....	.....	79.3	63.5
8-17-36	37	2-36	21.6	0.0	13:46	89.0	68.0	84.2	42.2	75.1	66.5	79.5	57.3
8-18-36	38	2-36	130.3	66.5	13:45	101.0	72.0	89.9	41.6	76.2	63.0	79.2	56.9
8-19-36	39	2-36	94.5	36.3	12:42	95.0	80.0	91.5	43.1	.....	.....	79.1	56.6
8-20-36	40	1-36	53.4	15.4	9:15	95.0	70.0	85.9	45.4	76.8	77.5	79.2	61.0
8-21-36	41	1-36	115.3	54.5	13:07	99.0	79.0	88.4	45.6	.....	.....	78.9	59.3
8-22-36	42	1-36	126.0	63.4	13:35	99.0	76.0	88.6	44.1	.....	.....	79.0	58.4
8-23-36	43	1-36	66.5	24.1	8:34	96.0	74.0	88.4	45.6	.....	.....	78.8	59.8
8-24-36	44	1-36	82.0	34.8	13:20	96.0	71.0	86.2	48.3	77.3	70.6	79.1	61.0
8-25-36	45	1-36	86.2	34.1	12:07	95.0	74.0	90.2	42.5	.....	.....	79.3	57.6
8-26-36	46	1-36	72.6	32.2	11:54	98.0	66.0	82.7	58.2	75.2	76.0	79.1	67.0
8-27-36	47	1-36	7.2	0.0	7:27	88.0	71.0	84.6	49.4	.....	.....	79.3	62.6
8-28-36	48	1-36	14.2	0.0	8:25	89.0	69.0	81.9	52.6	75.2	79.3	79.7	67.3
9-11-36	49	1-36	48.4	9.2	11:00	94.0	68.0	82.7	54.7	71.0	78.8	79.2	64.1
9 14-36	50	1-36	37.4	2.2	12:35	91.0	71.0	85.2	50.8	75.9	76.1	79.7	67.6
9-15-36	51	1-36	26.0	0.0	12:32	90.0	70.0	84.1	49.4	76.0	75.0	79.7	66.2

[*Continued on next page*]

the house was being operated with a constant dry bulb temperature of approximately 80 F. On the three days preceding this test the outdoor air temperature had attained a maximum value of 100 F, 102 F, and 105 F. During the greater portion of the early morning hours of July 7, 1936, the outdoor temperature was greater than 80 F and reached a minimum value of 78.5 F at 5 a.m. Since the outdoor air temperature was much greater than the indoor air temperature on the night previous to the test the windows remained closed and no outdoor air, other than the amount required for ventilation, was introduced into the house during the night. The air conditions at different locations are given in items 4 to 14 of Table 1. The dry bulb temperature on the second story was 2.2 F higher than that on the first story, indicating a slight unbalance in the cooling on the two stories. It was observed that this difference between the average temperatures of the first and second story rooms was of the order of only 0.5 F for a day in which the maximum outdoor temperature was 95 F, but increased to approximately 2.5 F for a day in which the maximum outdoor temperature was 108 F. This indicated that when the outdoor temperature was excessively high the cooling requirement for the second story became a relatively greater proportion of the total cooling requirement for both stories. The slight unbalance in the cooling load on the two stories during the few extremely hot days was not accompanied by any particular difficulties either in the conditions of operation or in the comfort maintained in the Residence. The temperature of the cooled air leaving the registers was approximately 68 F. The results obtained from tests made with three different arrangements of registers in the east bedroom are presented in a subsequent section of this paper. The rise in temperature of the air passing through the furnace casing and ducts was 3.7 F. The quantity of air circulated was 1265 cfm when the coils were dry. Increased frictional resistance when condensation was present on the coil surfaces, however, decreased this amount to 1231 cfm. Since part of the coil surfaces were always wet during actual operating periods, all calculations were based on 1231 cfm for the volume

Table 2—Complete Data and Results for Tests—Continued

Date 1936	Test No.	Series No.	Temperature Difference		Operation of the Fan						Overall Test Period		
					Night Air Cooling Previous to Test		Recirculation Previous to Test		Recirculation During Test Period		Start of Cooling with Water to Start of Night Air Cooling		
			Outdoor Max. to Average Indoor F	Average Outdoor to Avg. Indoor F	Duration Hours	Power Kw hr.	Duration Hours	Power Kw hr.	Duration Hours	Power Kw hr.	Start	End	Duration Hours
1	2	3	15	16	17	18	19	20	21	22	23	24	25
6-17-36	1	1-36	16.2	9.4	8.50	4.4	8.50	3.7	6.25	2.7	3:30p	9:45p	6.25
6-20-36	2	1-36	14.9	5.7	9.33	4.7	8.25	3.6	4.25	2.0	3:15p	7:30p	4.25
6-27-36	3	1-36	11.1	5.9	10.67	5.3	9.48	4.1	6.85	3.2	4:30p	11:30p	6.85
6-28-36	4	1-36	13.9	9.7	7.50	3.7	5.05	2.2	8.00	3.6	12:03p	8:03p	8.00
6-29-36	5	1-36	22.3	8.7	10.95	5.5	9.23	4.1	16.77	7.5	4:14p	9:00a	16.77
7- 2-36	7	1-36	12.8	5.8	12.67	6.5	11.08	4.8	1.92	0.7	6:05p	8:00p	1.92
7- 3-36	8	1-36	12.0	7.7	11.00	5.5	9.03	4.0	4.47	2.0	4:02p	8:30p	4.47
7- 4-36	9	1-36	21.0	9.4	10.50	5.2	4.25	1.8	21.75	9.6	11:15a	9:00a	21.75
7- 5-36	10	1-36	23.2	12.0	0.00	0.0	0.00	0.0	23.90	10.4	9:00a	9:00a	24.00
7- 6-36	11	1-36	25.5	12.1	0.00	0.0	0.00	0.0	24.00	10.7	9:00a	9:00a	24.00
7- 7-36	12	1-36	24.2	13.4	0.00	0.0	0.00	0.0	24.00	10.4	9:00a	9:00a	24.00
7- 8-36	13	1a-36	23.7	16.9	0.00	0.0	0.00	0.0	13.00	5.9	9:00a	10:00p	13.00
7- 9-36	14	1a-36	20.1	8.9	9.00	4.5	1.00	0.4	25.00	11.0	8:00a	9:00a	25.00
7-10-36	15	2-36	24.8	12.1	0.00	0.0	0.00	0.0	24.00	10.7	9:00a	9:00a	24.00
7-11-36	16	2-36	25.5	13.3	0.00	0.0	0.00	0.0	24.00	10.5	9:00a	9:00a	24.00
7-12-36	17	2-36	26.3	12.3	0.00	0.0	0.00	0.0	24.00	10.7	9:00a	9:00a	24.00
7-13-36	18	2-36	25.3	12.7	0.00	0.0	0.00	0.0	24.00	10.4	9:00a	9:00a	24.00
7-14-36	19	2-36	28.3	13.3	0.00	0.0	0.00	0.0	24.00	10.6	9:00a	9:00a	24.00
7-15-36	20	2-36	28.2	7.8	0.00	0.0	0.00	0.0	23.97	10.3	9:00a	9:00a	24.00
7-16-36	21	2-36	19.1	11.3	0.00	0.0	0.00	0.0	14.00	6.1	9:00a	11:00p	14.00
7-17-36	22	2-36	21.2	12.8	8.00	4.0	5.67	2.6	11.33	5.0	12:40p	12:00Mid.	11.33
7-18-36	23	2-36	12.9	4.4	7.00	3.2	8.32	3.6	4.68	2.1	3:19p	8:00p	4.68
7-23-36	24	2-36	17.6	13.0	9.90	4.9	4.68	2.0	11.57	5.1	11:41a	11:15p	11.57
7-24-36	25	2-36	12.8	4.1	7.75	3.8	7.47	3.2	5.03	2.1	2:28p	7:30p	5.03
7-25-36	26	2-36	10.7	5.3	11.50	5.7	9.98	4.3	6.35	2.8	4:59p	11:20p	6.35
7-26-36	27	2-36	19.7	8.7	7.67	3.8	2.18	1.0	23.82	10.2	9:11a	9:00a	23.82
7-27-36	28	2-36	21.3	13.0	0.00	0.0	0.00	0.0	10.67	4.6	9:00a	7:40p	10.67
7-28-36	29	2-36	13.5	8.3	11.33	5.6	7.47	3.1	6.53	2.9	2:28p	9:00p	6.53
8- 3-36	30	2-36	7.2	3.1	7.50	3.8	10.48	4.5	3.52	1.6	5:29p	9:00p	3.52
8- 9-36	31	2-36	13.7	6.8	7.00	3.7	9.72	4.2	3.78	1.7	4:43p	8:30p	3.78
8-10-36	32	2-36	7.5	4.4	9.98	4.7	8.88	3.8	3.45	1.6	3:53p	7:20p	3.45
8-13-36	33	2-36	17.1	9.8	12.00	6.2	6.75	3.0	10.00	4.6	1:45p	11:45p	10.00
8-14-36	34	2-36	10.4	3.1	7.25	3.5	8.67	3.8	17.33	7.6	3:40p	9:00a	17.33
8-15-36	35	2-36	16.1	5.4	0.00	0.0	0.00	0.0	24.00	10.4	9:00a	9:00a	24.00
8-16-36	36	2-36	15.7	8.9	0.00	0.0	0.00	0.0	13.60	6.1	9:00a	10:36p	13.60
8-17-36	37	2-36	9.5	4.7	8.40	4.3	9.43	4.1	5.57	2.3	4:26p	10:00p	5.57
8-18-36	38	2-36	21.8	10.7	9.00	4.7	6.20	2.7	19.80	8.6	1:12p	9:00a	19.80
8-19-36	39	2-36	15.9	12.4	0.00	0.0	0.00	0.0	13.30	5.9	9:00a	10:18a	13.30
8-20-36	40	1-36	15.8	6.7	9.45	4.9	4.98	2.1	20.27	8.9	12:44p	9:00a	20.27
8-21-36	41	1-36	20.1	9.5	0.00	0.0	0.00	0.0	24.00	10.5	9:00a	9:00a	24.00
8-22-36	42	1-36	20.0	9.6	0.00	0.0	0.00	0.0	24.00	10.6	9:00a	9:00a	24.00
8-23-36	43	1-36	17.2	9.6	0.00	0.0	0.00	0.0	14.00	6.1	9:00a	11:00p	14.00
8-24-36	44	1-36	16.9	7.1	8.00	4.0	4.52	2.0	21.48	9.5	11:31a	9:00a	21.48
8-25-36	45	1-36	15.7	10.9	0.00	0.0	0.00	0.0	12.33	5.4	9:00a	9:20p	12.33
8-26-36	46	1-36	18.9	3.6	9.60	4.9	6.85	3.0	19.15	8.3	1:51p	9:00a	19.15
8-27-36	47	1-36	8.7	5.3	0.00	0.0	6.52	2.9	5.48	2.5	3:31p	9:00p	5.48
8-28-36	48	1-36	9.3	2.2	10.00	4.9	9.32	4.1	2.68	1.3	4:19p	7:00p	2.68
9-11-36	49	1-36	14.8	3.5	11.00	5.5	12.50	5.3	4.33	2.0	7:30p	11:50p	4.33
9-14-36	50	1-36	11.3	5.5	11.50	5.6	8.37	3.7	6.13	2.7	3:22p	9:30p	6.13
9-15-36	51	1-36	10.3	4.4	9.50	4.6	9.65	4.1	4.85	2.1	4:39p	9:30p	4.85

of air circulated at a density of 0.0749 lb per cubic foot. The free area velocity of 700 fpm did not prove to be sufficient to carry any condensed moisture away from the surfaces of the cooling coil.

Heat Balance

Since the cooling coil was well lagged with corkboard to prevent heat gain, it was possible to obtain the actual cooling load either from the heat given up by the mixture of air and water vapor or from the heat absorbed by the water passing through the coil. However, the cooling studies made in 1935 had indicated that unless complete temperature traverses were made in the duct on both sides of the coil and proper corrections were applied to the averages of the temperatures obtained from the wet- and dry-bulb thermometers at the two reading stations on each side of the coil, the discrepancy in heat balance obtained by the two methods was very large. By applying proper corrections to the dry-bulb and wet-bulb reading, however, and using the corrected values of the latter to determine the enthalpy of the moist air entering and leaving the coil, it was possible to obtain a heat balance between the air and water within approximately 5 per cent. This method of calculation, however, does not show the distribution between sensible and latent heat loads. This distribution was obtained, as shown in Item 21 Table 1, by using the corrected values for the dry-bulb temperatures, and by multiplying the weighed amount of condensation from the coil by a constant representing the heat given up by the change in moisture content of the air per pound of water vapor condensed. This latter included the latent heat and superheat in the water vapor condensed, and the change in the superheat of the water vapor remaining in the air after passing the cooling coil, and amounted to 1,050 Btu per pound of vapor condensed. The total of the heat given up by the air as computed by this method was within 4 per cent of the calculated heat absorbed by the water. For the purpose of analysis, and in the general results shown in Table 2, the cooling load calculated from the water circulated through the coil, as

Table 2—Complete Data and Results for Tests—Continued

Date 1936	Test No.	Series No.	Water Circulation Through Cooling Coil		Water Circulation Through Cooling Coil		Total Moisture Condensed Lb	Heat Absorbed by Cooling Coil During Overall Test Period, Btu			Ratio Moisture to Total Load
			Time in Hours	Quantity in Gallons	Temp. at Inlet F	Temp. Rise Through Coil F		Load Due to Moisture	Sensible	Total	
1	2	3	26	27	28	29	30	31	32	33	34
6-17-36	1	1-36	3.08	778	52.1	9.6	0.00	0	62,300	62,300	0.000
6-20-36	2	1-36	1.95	473	51.6	10.1	0.00	0	39,800	39,800	0.000
6-27-36	3	1-36	4.35	1099	51.2	10.6	4.19	4,400	92,700	97,100	0.045
6-28-36	4	1-36	6.80	1463	52.0	10.5	1.01	1,060	126,940	128,000	0.008
6-29-36	5	1-36	8.25	2082	52.4	10.4	3.53	3,710	176,730	180,440	0.021
7- 2-36	7	1-36	0.83	207	52.2	10.3	0.00	0	17,750	17,750	0.000
7- 3-36	8	1-36	2.42	607	52.5	10.4	0.00	0	52,550	52,550	0.000
7- 4-36	9	1-36	12.55	3155	52.7	10.7	15.77	16,550	265,030	281,580	0.059
7- 5-36	10	1-36	19.53	4950	53.2	10.3	26.82	28,150	396,650	424,800	0.066
7- 6-36	11	1-36	20.18	5025	52.6	11.2	39.64	41,630	427,870	469,500	0.089
7- 7-36	12	1-36	21.48	5457	51.7	11.4	45.49	47,760	470,480	518,240	0.092
7- 8-36	13	1a-36	13.00	3955	52.1	10.0	29.44	30,920	298,530	329,450	0.094
7- 9-36	14	1a-36	17.87	5452	52.1	9.7	50.16	52,670	387,870	440,540	0.120
7-10-36	15	2-36	19.35	3635	47.4	16.0	49.21	51,670	432,800	484,470	0.107
7-11-36	16	2-36	21.33	3460	46.9	18.0	57.40	60,300	459,200	519,500	0.116
7-12-36	17	2-36	20.93	3379	47.0	18.0	49.53	52,000	455,500	507,500	0.103
7-13-36	18	2-36	21.02	3390	46.7	18.5	62.28	65,400	457,600	523,000	0.125
7-14-36	19	2-36	20.80	3397	46.4	18.2	42.44	46,570	470,930	515,500	0.086
7-15-36	20	2-36	16.05	2638	46.7	17.8	62.09	65,200	326,440	391,640	0.167
7-16-36	21	2-36	12.55	2071	46.8	17.7	52.72	55,360	250,050	305,410	0.181
7-17-36	22	2-36	9.70	1584	45.5	17.9	9.55	10,030	226,470	236,500	0.042
7-18-36	23	2-36	2.68	433	49.4	16.2	6.07	6,370	52,120	58,490	0.109
7-23-36	24	2-36	8.17	1338	46.2	18.2	19.92	20,920	182,180	203.100	0.103
7-24-36	25	2-36	1.22	197	46.1	17.3	0.33	350	28,090	28,440	0.012
7-25-36	26	2-36	2.18	358	47.6	19.5	10.60	11,130	47,170	58,300	0.191
7-26-36	27	2-36	13.15	2150	46.4	18.6	56.25	59,050	274,550	333,600	0.177
7-27-36	28	2-36	9.65	1575	46.3	18.7	38.78	40,720	204,780	245,500	0.166
7-28-36	29	2-36	3.95	647	46.3	18.5	9.44	9,910	89,890	99,800	0.099
8- 3-36	30	2-36	1.20	196	45.9	18.1	0.62	650	29,000	29,650	0.022
8- 9-36	31	2-36	1.18	194	46.2	18.0	0.73	770	28,370	29,140	0.026
8-10-36	32	2-36	1.31	215	46.3	18.2	1.54	1,620	31,070	32,690	0.050
8-13-36	33	2-36	7.28	1186	46.1	17.9	26.36	27,690	149,410	177,100	0.156
8-14-36	34	2-36	2.15	348	46.4	19.3	4.02	4,220	51,900	56,120	0.075
8-15-36	35	2-36	9.80	1601	47.0	18.0	47.10	49,440	191,060	240,500	0.206
8-16-36	36	2-36	9.75	1490	46.6	18.4	39.15	41,100	187,800	228,900	0.180
8-17-36	37	2-36	1.85	301	45.8	17.8	1.30	1,370	43,330	44,700	0.031
8-18-36	38	2-36	11.83	1930	46.1	18.1	32.95	34,600	254,800	280,400	0.120
8-19-36	39	2-36	11.17	1818	46.2	18.1	44.69	46,930	227,560	274,490	0.171
8-20-36	40	1-36	7.41	1980	52.2	11.2	23.95	25,150	159,750	184,900	0.136
8-21-36	41	1-36	14.18	3589	52.0	10.9	45.26	47,525	278,745	326,370	0.146
8-22-36	42	1-36	14.75	3732	52.1	11.1	47.74	50,125	295,350	345,475	0.145
8-23-36	43	1-36	9.82	2486	51.8	11.1	33.98	35,680	194,500	230,180	0.155
8-24-36	44	1-36	10.10	2570	52.2	11.3	35.20	36,960	205,245	242,205	0.153
8-25-36	45	1-36	10.33	2621	52.0	11.2	37.52	39,395	205,380	244,775	0.161
8-26-36	46	1-36	7.05	1796	51.9	11.1	32.11	33,715	132,520	166,235	0.203
8-27-36	47	1-36	1.60	410	52.2	11.4	1.56	1,640	37,290	38,930	0.042
8-28-36	48	1-36	0.95	243	52.4	11.9	0.96	1,010	23,050	24,060	0.042
9-11-36	49	1-36	0.90	229	.....	.....	0.24	250	20,340	20,590	0.012
9-14-36	50	1-36	1.50	382	52.0	11.9	0.94	990	36,890	37,880	0.026
9-15-36	51	1-36	0.98	250	52.2	11.7	0.30	315	24,060	24,375	0.013

shown in Item 26, Table 1, was accepted as correct. The latent heat load was calculated from the weighed amount of moisture condensed from the air and the sensible heat load was obtained by difference. As shown in Item 21 in Table 1, the moisture load for the day under consideration was 9.5 per cent of the total load. It may also be observed from column 34 in Table 2 that for the previous to this test and the cooling plant had been operating intermittently to maintain a constant dry-bulb temperature in the rooms both day and night. On this test the plant operated intermittently from 2:50 a.m. to 7:20 a.m. and continuously from 7:20 a.m. to 3:50 a.m. maintaining the dry-bulb temperature between 78 F and 81.5 F.

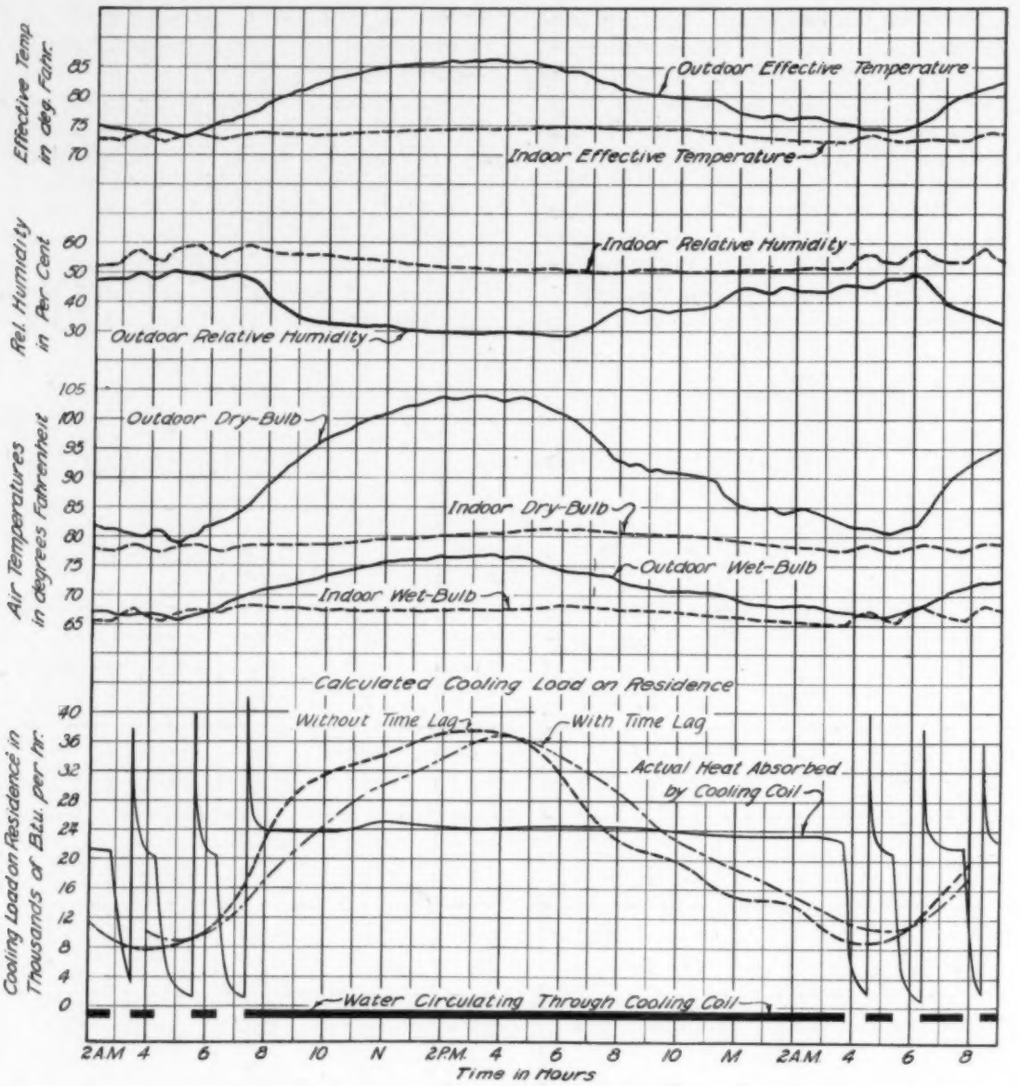

Fig. 4—Actual and calculated cooling load on residence and air temperatures. Test No. 12, Series 1-36, July 7, 1936. 52 F cooling water

tests made during the season of 1936 the moisture load varied from 0 to 21 per cent of the total load. A more detailed discussion of this ratio of moisture load to the total load is presented in a subsequent section of this paper.

Operating Characteristics

The operating characteristics with 52 F cooling water and with a constant dry-bulb temperature in the rooms of approximately 80 F are shown in Fig. 4, for a day on which the outdoor temperature rose to a maximum of 104 F. The house had remained closed for several days

At 5:35 a.m. after the water circulation had been stopped about 80 min, the wet-bulb temperature was 67.8 F corresponding to a relative humidity of 59 per cent. When the water circulation was started the relative humidity decreased to 56 per cent, but increased again to 59 per cent almost immediately after the water circulation was stopped. This alternate decrease and increase of relative humidity in the house was characteristic of the intermittent operation of the plant.

It is of interest to note that in the summer of 1935 when the outdoor conditions were such that the air contained more moisture than it did in 1936, which was an excessively dry season, the total variation in the indoor

relative humidity during the on- and off-periods of plant operation was comparatively large, amounting to approximately 13 per cent. It is probable that if the 1935 season had been more normal from the standpoint of outdoor humidity, the magnitude of the variation in indoor humidity, resulting from intermittent operation of the plant, would have been much smaller than 13 per cent, but it is hardly probable that it would have been as small as the 3 per cent shown in Fig. 4. It is also possible that some of the difficulties that were experienced in 1935 when the cooling plant was operated intermittently in order to maintain a constant dry-bulb temperature might have been avoided if the outdoor air had contained less moisture. It is apparent that the indoor relative humidity that exists during the off-periods of any intermittently operated cooling unit is largely dependent on the prevailing outdoor humidity conditions. During the season of 1936 because of the very small changes in the relative humidity occurring between the on-periods and the off-periods of the cooling plant, and resulting as a consequence of the excessively dry season, the conditions maintained in the house were entirely satisfactory even for those days on which the plant was operating intermittently over a large portion of the time.

When the plant operated continuously, as from 7:20 a.m. to 3:50 a.m., the relative humidity stabilized at between 50 and 55 per cent, representing an effective temperature of approximately 74 deg. During the period between 4 p.m. and 8 p.m. when the average indoor dry-bulb temperature was approximately 81 F, the relative humidity in the house was approximately 51 per cent and the corresponding effective temperature was 74.5 deg. In spite of the fact that on these very hot days the capacity of the cooling plant was not quite sufficient to prevent the indoor temperatures from exceeding 80 F, particularly on the second story, the effective temperature did not exceed 75 deg at any time. Hence the indoor conditions, both during the periods of intermittent operation and during the periods of continuous operation of the cooling plant, were on the upper borderline for comfort and could be regarded as entirely satisfactory. As indicated from the values listed in column 2 of Table 1, the indoor conditions that were maintained with the water entering the cooling coil at 46.9 F were practically identical with those maintained when the temperature of the entering water was 51.9 F. That is, in both series of tests, the indoor dry-bulb temperature, the indoor relative humidity, and the indoor effective temperature respectively were maintained at approximately the same values, irrespective of the temperature of the water entering the coil.

From the top set of curves in Fig. 4 it may be observed that on this day, in common with a number of other similar days occurring during the summer of 1936, the effective temperature of the outdoor air was always greater than the effective temperature of the indoor air, Hence no cooling could be effected during the night by the introduction of large quantities of outdoor air. However, this limitation during the extremely hot portion of the summer did not preclude the effective use of outdoor air at night during the greater portion of the season when the maximum outdoor temperature did not exceed approximately 100 F.

Acknowledgments

The results presented in this paper were obtained in connection with the summer cooling investigation (1936) in the Research Residence‡ at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to M. K. Fahnestock, Research Assistant Professor, and R. B. Engdahl, Research Assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

‡The Research Residence in Urbana, Ill., was built, furnished, and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

(Part 2 to be published in June, 1937)

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Heat Insulation Tests and Their Application

By E. A. Allcut* and F. G. Ewens** *(MEMBERS)*
Toronto, Ont., Can.

WHEN the published results of tests made on heat insulating materials by the hot plate method are compared, considerable differences are found. Some of these discrepancies are probably due to differences in the structure of materials which are known by the same name but are not made at the same time or by the same process. Further, the information given in the tables of conductivities and conductances is generally incomplete, in that the size and thickness of specimen tested, the moisture present in the specimen, and the temperature difference between the hot and cold plates, are not stated. The attention of the authors was directed to these questions by two circumstances:

(1) Tests made by them on fibrous materials (e.g. rock wool, etc.) gave conductivities which were not only considerably higher than the published figures, but varied with the temperature difference used. This is illustrated by Curve A, Fig. 1, which indicates the conductivities obtained with temperature differences between the hot and cold plates varying from 29 to 89 F. In this case, some of the increase was due to rising mean temperature, as the cold plate was cooled by water at a temperature that was approximately constant in all of these tests. The change of conductivity, however, was too great to be entirely accounted for by this explanation and it appeared, therefore, that the increased temperature difference was a contributory factor.

(2) Tests made by the authors, and many others, indicated that the conductivity, calculated in the usual way from the conductance, increased considerably with the thickness of the specimen in the case of some materials, notably fibre boards as shown in Table 1 and Fig. 2. This was thought to be due to surface resistance to the passage of heat, and temperature gradients obtained by means of thermocouples cast inside specimens 2½ in. thick appeared to confirm this view, as illustrated in Fig. 3.

If this surface resistance is appreciable in extent, the heat transmission per inch, as calculated from the conductance of the specimen, is not the true conductivity of the material. Conductivity is a characteristic of the *material,* while conductance is a characteristic of the *specimen.* The authors prefer, therefore, to use the expression *apparent conductivity* for this property.

Calibration and Checking

When the results recorded were obtained, they were considered to be due either to some fault in the apparatus or to improper technique. The hot plate used was 24-in. square, with a guard ring 4-in. wide, and was surrounded by 3½-in. of cork board to reduce edge losses. Five thermocouples, arranged as shown in Fig. 4, were mounted on each of the hot and cold plates, and on each hot plate there were also four couples (marked *G T*) to indicate the guard ring temperatures. These thermocouples were all mounted on sheets of paper and were in direct contact with the specimens but were not in contact with the metal plates. The potential difference between the two junctions of each thermocouple

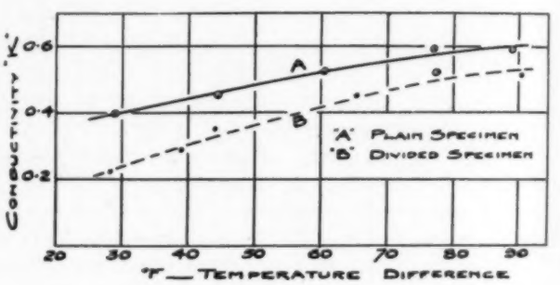

Fig. 1—Conductivities of plain and divided specimens (as shown in Fig. 7) indicating influence of convection

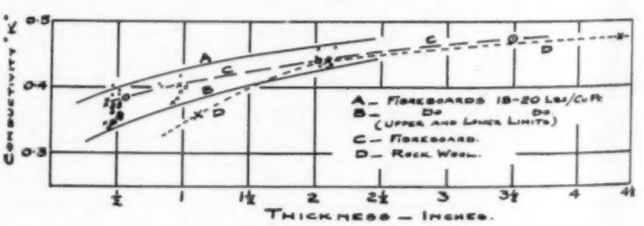

Fig. 2—Effect of thickness on (apparent) conductivities

was measured by a potentiometer, and the temperature of the common cold junction was indicated by a thermometer. The same potentiometer was used to indicate the current flowing through the central part of the hot plate by means of the potential drop across a standard resistance. The heat passing through the central part of the specimen was then calculated from the I^2R loss in the heating element.

The following possible sources of error were investigated:

1. *Measurement of electrical input*
   a. Accuracy of potentiometer.
   b. Resistance of heating coil.
   c. Standard resistance.

These were checked by using an ammeter and voltmeter in place of the potentiometer and additional readings were taken

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with a standard wattmeter. The accuracy of the temperature indications, which were checked separately, also confirmed the potentiometer readings. The resistances of the heating coil, guard ring windings and standard resistance were checked by independent observers. All of these observations confirmed the accuracy of the results obtained on this apparatus.

2. *The distribution of heat and edge losses*

A survey of the rate of heat loss for the central plate and guard ring windings indicated some variation in the watts per square foot supplied to the different parts of the plate. This was corrected and some of the tests repeated. but the difference in observed conductance was negligible (about 2 per cent). Further, the I^2R loss for the whole plate, including guard ring, was measured in several tests and the calculated conductances, for a simple hot plate with edge insulation, were almost exactly the same as those previously obtained from the centre winding. The edge losses were unlikely to be large in any case, as, in the later tests, the mean temperatures of the specimens were within a few degrees of the atmospheric temperature. A further check was provided by tests made on cork boards of thicknesses up to 4 in., as given in Table 1, which agreed closely with the known characteristics of cork board.

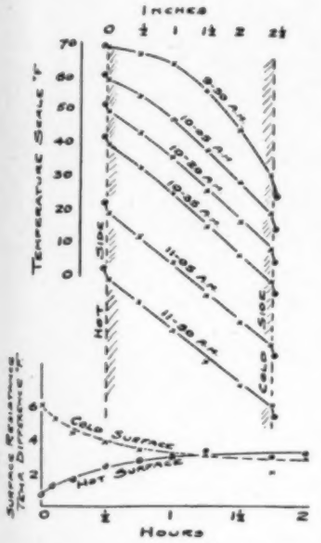

Fig. 3—Temperature gradients and surface resistances of special concrete

3. *Temperatures*

A rough check on the temperatures of the cold plates was obtained by comparing the water, or brine temperatures with those given by the thermocouples, and these were in reasonable agreement. In view of the fact that there were 18 thermocouples on the two hot plates and 10 on the two cold plates, any error must be of a common nature, because individual errors would be at once apparent. Calibrated couples were therefore inserted on the hot side and were kept entirely separated from the others, but no appreciable differences were detected. Another couple was inserted to read the temperature differences only between the hot and cold sides, and these agreed with the other readings. Also, tests were made with thermocouples embedded in the materials themselves, and the values of k were calculated from the temperature gradients, Table 2, with the same results as before.

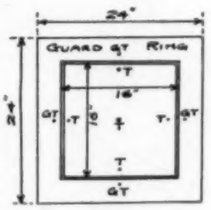

Fig. 4 — Arrangement of thermocouples in 24-in. hot plate and guard ring

4. *Surfaces*

It was considered that the method of preparing specimens of rock wool, etc., by packing them in a wooden frame to the required density and then forming the front and back of the specimen by means of thin plywood, might introduce some errors. Accordingly, the tests were repeated with wire mesh in place of the plywood, but no error was discovered. In the former case, corrections were made for the resistance of the plywood.

Size of the Plate

Tests previously made, shown in Fig. 5, indicated that with ½-in. fibre boards, the same results were obtained whether an 8-in. square guarded hot plate, or a 24-in. plate, were employed. Subsequent tests on 1-in. cork board, Tables 1 and 3, indicated the same apparent conductivities on both the large and small plates, but when

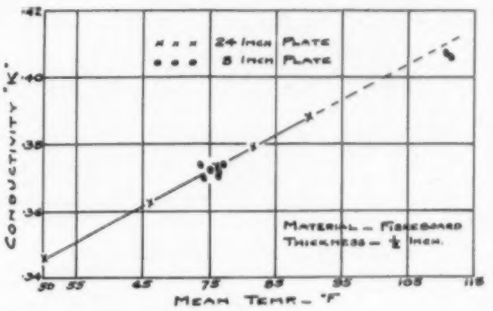

Fig. 5—Values of k for ½-in. fibre board from 8-in. and 24-in. hot plates

thicker specimens were used in the large plate, material differences were observed. Table 3 shows the comparative results obtained on the two sizes when both area and thickness were increased. The influence of area alone could not be observed at thicknesses exceeding 1-in. because the edge losses from the 8-in. plate would probably be excessive if thicker specimens were used. It is interesting to note that the apparent conductivities of rock wool, glass wool, aluminum foil and fibre board, as obtained on the small plate, agree fairly closely with the values usually quoted for these materials, but those on the larger and thicker specimens are considerably higher. It is possible that in these tests there is an analagous condition to that which is obtained in tensile tests on metals, where the form and size of the test piece have a considerable influence on the test results obtained and that, in heat transmission tests, the ratio of thickness to area should be standardized.

Thickness

These observations led to an investigation of the influence of thickness alone on the apparent conductivity as given by the 24-in. plate. Characteristic values are recorded in Table 1, and it is interesting to note that the value of k for cork board remains practically constant at all thicknesses, from 1½ to 4 in. The higher value for the 1-in. thickness is probably fortuitous. However, all of the other materials give increasing values of k as the thickness increases and representative test figures on fibre boards from a number of different laboratories are given in Fig. 2, so that this condition is general. In the case of fibre boards, this might be accounted for by the decreasing effect of surface resistance as the thickness becomes greater. The increasing conductivities in the case of packed materials such as rock wool and glass wool may be due either to surface effects or to convection currents, of which some evidence is given later on in this paper.

Table 1—Effect of Thickness, 24-in. Hot Plate

Corkboard					
Thickness, in.	4.04	3.00	2.00	1.49	1.00
Density, lb–cu ft.	8.27	9.23	7.74	9.55	8.35
Mean Temp. F.	64.77	63.28	53.40	56.80	56.44
Temp. Diff. F.	33.83	31.20	29.50	27.28	39.64
Conductance, C	0.0755	0.101	0.152	0.203	0.333
Conductivity, k	0.304	0.303	0.304	0.303	0.333
Rockwool A				**Glass Wool**	
Thickness, in.	4.375	2.09	1.125	2.00	4.06
Density, lb–cu ft.	10	9.55	10	1.50	1.50
Mean Temp. F.	72.9	73.0	60.0	74.8	79.09
Temp. Diff. F.	32.56	39.18	31.1	41.95	49.09
Conductance, C	0.109	0.211	0.315	0.2065	0.1308
Conductivity, k	0.478	0.442	0.355	0.413	0.532
Fibre Board					
Thickness, in.	0.554	3.551	0.554	2.03	
Density, lb–cu ft.	15.50	15.60	15.50	15.0	
Mean Temp. F.	76.4	73.1	80.72	79.34	.....
Temp. Diff. F.	26.4	30.36	57.00	50.18	
Conductance, C	0.688	0.1338	0.695	0.215	
Conductivity, k	0.381	0.475	0.384	0.436	

Temperature Difference

A suggestion was made in connection with some of the early tests, that the temperature differences between the hot and cold plates were not great enough to give reliable results. Accordingly, these were increased and higher apparent conductivities were obtained in tests made on the 24-in. plate.

The effects of increasing temperature differences with the 8-in. plate are shown in Table 4, and even in these tests with varying mean temperature, no great differences in the value of k were observed. Additional tests made on the 24-in. plate with ½-in. fibre board, as indicated in Table 5, confirmed previous experience with specimens of that thickness as detailed in a previous paper by one of the authors.[1]

The following figures were then published:

Mean Temperature, F.	61.5	58.1	59.8
Temp. Difference, F.	22.45	48.60	60.20
Conductivity, k	0.384	0.377	0.382

[1]Heat Insulation as Applied to Buildings and Structures, E. A. Allcut, *Proc., Inst. Mech. Engrs.*, Vol. 128, 1934, p. 195.

The same observation applies to two tests made on a special concrete (2-in. thick) as indicated in Table 5. Tests of other materials on the 24-in. plate, however, including fibre board, cork board, glass wool, rock wool, etc., gave increasing values of k as the temperature difference was raised as presented in Table 5 and Fig. 6. It is noticeable that the results obtained were fairly consistent at temperature differences below 70 F, but at higher values they became somewhat erratic. A peculiar fact was that at these higher temperature differences the direction of temperature change seemed to affect the results. Thus, if the change was from a lower to a higher temperature difference, higher results were obtained than when the change was in the opposite direction. This is indicated by arrows on Table 5 and Fig. 6, and it might be assumed that insufficient time was allowed for equilibrium to be set up before each test was started, *but in every case no test readings were taken until the conditions were absolutely steady*, and in many instances several days elapsed before the first set of readings was taken. There seems to be some kind of hysteresis effect, however.

Convection Currents

It was thought that the high results obtained with the larger and thicker specimens of packed fibrous materials might be due to the circulation of air within the material. The tests already made on rock wool, Table 6A and Curve *A*, Fig. 1, packed to a density of 10 lb per cubic foot were repeated on the 24-in. hot plate with the same specimens (4-in. thick). In this case, however, the specimens were divided into 12 horizontal sections by pieces of paper 2-in. apart, illustrated in Fig. 7, and the results obtained are given in Table 6B and Curve *B*, Fig. 1. It is worthy of note that, under these conditions, the value of k obtained with a temperature difference of 40 F is very close to those obtained on the 8-in. plate. Also that the Curve *B* lies considerably below Curve *A* and is nearly parallel to it.

An attempt was made to obtain further evidence from thermocouples placed inside the specimens themselves. Accordingly, a box of rock wool packed to 10 lb per cubic foot was provided with a row of thermocouples

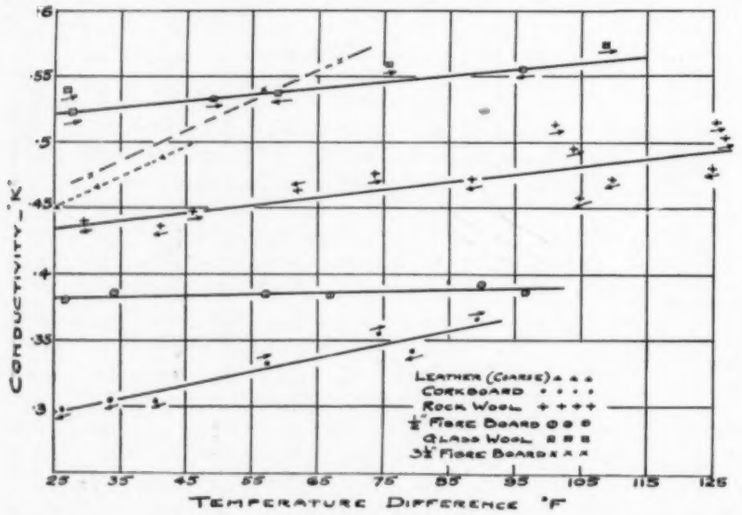

Fig. 6—Effect of temperature difference between hot and cold sides of specimen on values of k

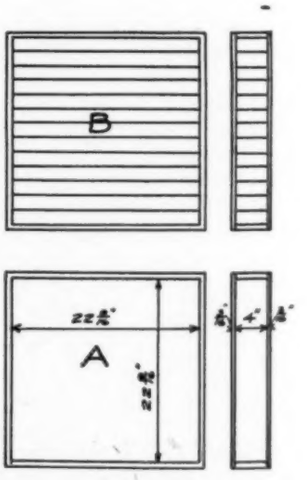

Fig. 7—Plain and divided specimens for packed materials (e.g. rock wool)

Table 2—Apparent Conductivities of Rock Wool—24 In. Plate

Specimen 4 in. thick in wooden box with plywood sides

Test No.	Outside Thermocouples			Inside Thermocouples		
	Mean Temp. F	Temp. Diff. F	k	Mean Temp. F	Temp. Diff. F	k
11 A	60.0	29.0	0.398	60.3	27.27	0.405
11 E	71.2	44.38	0.455	71.5	41.33	0.456
11 B	74.6	60.21	0.528	75.3	54.83	0.535
11 D	84.2	77.17	0.518	83.0	70.66	0.512
11 F	87.1	76.72	0.593	87.1	67.76	0.608
11 C	91.1	89.10	0.590	91.9	82.57	0.593

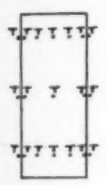

Fig. 8 — Arrangement of thermocouples in packed specimens

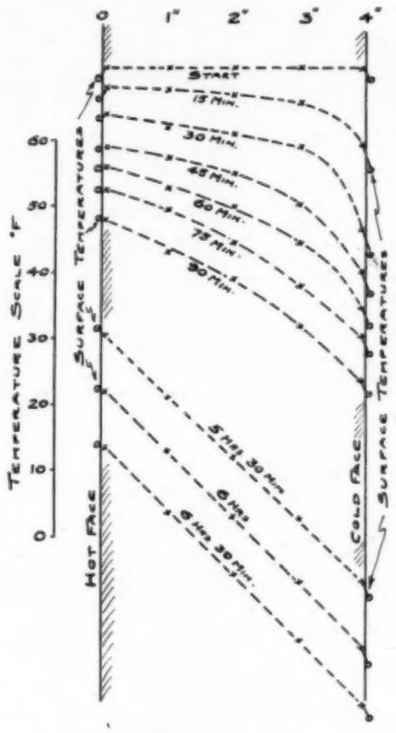

Fig. 9 — Temperature gradients at different times obtained from thermocouples, as shown in Fig. 8

at the top, middle and bottom, respectively, of the measured (central) area. Those at the top and bottom were 1 in. apart, and in the middle 2 in. apart as illustrated in Fig. 8. Readings of these were taken at 15-min intervals for the first 4 hours, and at longer intervals subsequently, until equilibrium was established. Some examples are given in Fig. 9, showing that the temperature gradient ultimately became a straight line, or very nearly so.

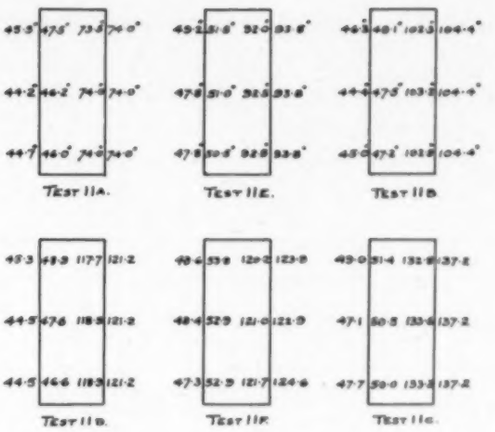

Fig. 10—Temperatures on surface and inside specimens of rock wool

The temperatures of the outside and inside of the specimen, with varying temperature differences, after steady conditions had been established, are shown in Fig. 10. It will be noted that in practically all cases, the inside temperature on the hot side decreases from the bottom to the top, and that the reverse is the case on the cold side. Variations in the outside temperatures do not appear to account for this and it is probable that, even with a high packing density, some air circulation takes place, and this may account for some of the phenomena described previously.

This also provided a useful check on the accuracy of the previous work as reported in Table 2. Calculations for k were made in this table from the readings of the outside thermocouples in the usual way, and further calculations were made from the temperature gradients inside the specimens.

Table 3—Influence of Size of Plate and Varying Thickness

Material	Rockwool B Bats		Rockwool B Loose		Crumpled Aluminum Foil		Glass Wool		Corkboard		Rockwool A	
Plate	8 In.	24 In.	8 In.	24 In.	8 In. 3 Sheets	24 In. 6 Sheets	8 In.	24 In.	8 In.	24 In.	8 In.	24 In.
Thickness, in.	1.10	2.00	1.00	2.06	1.00	2.06	1.00	2.00	1.00	2.00	1.00	2.09
Density, lb–cu ft	3.72	3.60	6.00	6.00	.....	.....	1.50	1.50	8.35	7.74	12.10	9.55
Mean Temp. F	81.8	79.0	85.6	79.0	83.8	79.4	80.4	74.8	76.8	53.4	78.3	73.0
Temp. Diff. F	41.58	49.47	41.52	47.95	37.26	48.14	42.80	41.95	30.87	29.50	29.80	39.18
Conductance, C	0.256	0.1755	0.262	0.180	0.321	0.256	0.285	0.2065	0.330	0.152	0.277	0.211
Conductivity, k	0.282	0.351	0.262	0.372	0.321	0.528	0.285	0.413	0.330	0.304	0.277	0.442

Material	Fibre Board		Leather Shavings Coarse		Leather Shavings Fine		Sawdust	
Plate	8 In.	24 In.	8 In.	24 In.	8 In.	24 In.	8 In.	24 In.
Thickness, in.	1.02	2.03	1.00	4.00	1.00	4.06	1.00	4.00
Density, lb–cu ft	16.95	15.00	9.25	10.00	13.23	13.00	12.00	12.00
Mean Temp. F	83.4	79.3	80.5	71.3	74.66	65.8	87.5	79.5
Temp. Diff. F	40.62	50.18	36.40	38.16	34.85	32.14	41.20	43.50
Conductance, C	0.345	0.215	0.268	0.122	0.309	0.143	0.368	0.158
Conductivity, k	0.352	0.436	0.268	0.488	0.309	0.585	0.368	0.632

Note: All *conductances* are expressed in Btu per square foot per degree Fahrenheit per hour (C).
All *conductivities* are expressed in Btu per square foot per degree Fahrenheit per inch thickness per hour (k).

Table 4—Tests of Rock Wool on 8-In. Guarded Hot Plate

Size of Specimens 8 in. x 8 in. x 1 in.

Density 12.1 lb per cubic foot

Supported in thin timber frame with paper sides.

Mean Temp. F	78.3	81.1	87.5	101.4
Temp. Diff. F	29.8	54.7	54.9	82.6
Conductivity, *k*	0.277	0.269	0.263	0.285

Table 5—Effects of Temperature Difference—24-In. Plate

FIBRE BOARD

Thickness, in.	0.554	0.554	0.554	0.554	0.554	0.554
Density, lb-cu ft	15.50	15.50	15.50	15.50	15.50	15.50
Mean Temp. F	76.42	80.29	80.72	79.30	82.10	83.48
Temp. Diff. F	26.40	34.07	57.00	66.90	89.87	96.50
Conductance, *C*	0.688	0.697	0.695	0.692	0.708	0.698
Conductivity, *k*	0.381	0.386	0.384	0.383	0.392	0.386

FIBRE BOARD

Thickness, in.	3.551	3.551	3.551
Density, lb-cu ft	15.6	15.6	15.6
Mean Temp. F	73.11	78.68	75.91
Temp. Diff. F	30.36	56.54	68.29
Conductance, *C*	0.1338	0.1521	0.1581
Conductivity, *k*	0.475	0.540	0.561

CONCRETE (SPECIAL)

Thickness, in.	2.175	2.175
Density, lb-cu ft	93.75	93.75
Mean Temp. F	69.7	71.6
Temp. Diff. F	19.23	31.92
Conductance, *C*	1.278	1.273
Conductivity, *k*	2.778	2.767

CORK BOARD

Thickness, in.	4.04	4.04	4.04	4.04	4.04	4.04
Density, lb-cu ft	8.27	8.27	8.27	8.27	8.27	8.27
Mean Temp. F	64.77	66.68	65.01	66.74	65.10	65.35
Temp. Diff. F	33.83	79.10	57.26	40.12	74.43	25.94
Conductance, *C*	0.0755	0.0846	0.0822	0.0755	0.0895	0.07
Conductivity, *k*	0.304	0.342	0.332	0.304	0.361	0.297
	→	←	→	←	→	←

GLASS WOOL

Thickness, in.	4.06	4.06	4.06	4.06	4.06	4.06	4.06
Density, lb-cu ft	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Mean Temp. F	73.70	76.80	79.09	79.70	80.95	82.06	80.98
Temp. Diff. F	27.57	26.84	49.04	75.58	108.80	96.11	58.75
Conductance, *C*	0.1285	0.1325	0.1308	0.1374	0.1415	0.1367	0.1320
Conductivity, *k*	0.523	0.540	0.532	0.560	0.575	0.556	0.537
	→	→	→	→	→←	←	

GLASS WOOL

Thickness, in.	2.06	2.00
Density, lb-cu ft	1.92	1.50
Mean Temp. F	75.5	74.8
Temp. Diff. F	27.14	41.95
Conductance, *C*	0.194	0.2065
Conductivity, *k*	0.400	0.413

ALUMINUM FOIL (CRUMPLED)

Thickness, in.	2.06	2.06	2.06
Density, lb-cu ft	Three	Sheets per inch	
Mean Temp. F	68.8	72.1	79.4
Temp. Diff. F	29.88	35.67	48.14
Conductance, *C*	0.217	0.241	0.256

ROCK WOOL *A*

Thickness, in.	2.06	2.09	
Density, lb-cu ft	10.0	9.55	
Mean Temp. F	72.5	73.0	
Temp. Diff. F	24.01	39.18	
Conductance, *C*	0.176	0.211	
Conductivity, *k*	0.363	0.442	

Table 5—continued

ROCK WOOL *A*

Thickness, in.	4.00	4.00	4.00	4.00	4.00	4.00	4.00
Density, lb-cu ft	10	10	10	10	10	10	10
Mean Temp. F	75.60	79.68	82.71	84.06	83.40	83.93	81.71
Temp. Diff. F	46.00	73.58	100.99	127.21	124.87	125.24	109.74
Conductance, *C*	0.1117	0.1190	0.1282	0.1257	0.1200	0.1285	0.1180
Conductivity, *k*	0.447	0.476	0.513	0.503	0.480	0.514	0.472
	→	→	→	→←		→←	

ROCK WOOL *A*—(CON.)

Thickness, in.	4.00	4.00	4.00	4.00	4.00	4.00
Density, lb-cu ft	10	10	10	10	10	10
Mean Temp. F	78.50	81.62	82.54	80.25	79.25	81.40
Temp. Diff. F	104.85	88.62	61.77	40.82	29.19	103.73
Conductance, *C*	0.1143	0.1180	0.1158	0.1090	0.1100	0.1240
Conductivity, *k*	0.457	0.472	0.463	0.436	0.440	0.496
	←	←	←	←	←	→

→ *Temp. difference increasing*
← *Temp. difference decreasing*

LEATHER SHAVINGS (COARSE)

Thickness, in.	4.0	4.0	4.0
Density, lb-cu ft	10.0	10.0	10.0
Mean Temp. F	66.8	68.6	71.3
Temp. Diff. F	30.14	33.22	38.16
Conductance, *C*	0.113	0.1165	0.122
Conductivity, *k*	0.452	0.466	0.488

Table 6A—Tests of Specimen Packed in Wooden Box with Plywood Sides

Material—Rock Wool. Packing Density 10 lb per cu ft

Mean Temp. F	59.74	71.21	74.63	87.08	84.25	91.11
Temp. Diff. F	29.00	44.38	60.21	76.72	77.17	89.10
Conductance, *C*	0.0994	0.1137	0.1320	0.1482	0.1295	0.1475
Conductivity, *k*	0.397	0.455	0.528	0.593	0.518	0.590

Table 6B—Tests of Specimen Packed in Wooden Box with Plywood Sides

Divided into horizontal layers 2-in. thick by means of paper partitions.

Mean Temp. F	54.33	58.88	63.30	71.63	85.14
Temp. Diff. F	27.93	38.43	43.70	65.44	90.04
Conductance, *C*	0.0548	0.0710	0.0907	0.1119	0.1270
Conductivity, *k*	0.219	0.284	0.363	0.448	0.508

Table 7—Percentage of Moisture Present in Insulating Materials

MATERIAL	THICKNESS, IN.	MOISTURE, PER CENT
Corkboard	3.0	0.80
Corkboard	2.0	1.03
Corkboard	1.5	1.00
Corkboard	1.0	1.27
Rockwool, *A*	....	0.57
Fibreboard	3.5	4.9
	0.5	4.7
Concrete (special)	2.125	3.3
Leather (coarse)	....	12.5
Leather (fine)	....	14.4
Sawdust	....	8.2

Table 8—Percentage Increase in Weight of Insulating Materials

MATERIAL	CORKBOARD (1 IN. THICK)	GLASS WOOL	ROCK WOOL A	ROCK WOOL B	FIBRE BOARD (½ IN. THICK)
Water absorbed after 72 hours, per cent by weight	1.05	0.4	0	0	10.3

Table 9—Moisture Absorption by Fibre Board Specimens 8-In. Square and Various Thicknesses Exposed to Air at 65 F and 85-90 Per Cent Relative Humidity

Sample No.	Thickness In.	Make	Density Lb/cu ft as Received	Dry Weight Grammes	Moisture as Received Per Cent	Final Moisture		Moisture After Air Drying for 50 Hours	
						Per Cent	Grammes per cu in.	Per Cent	Grammes per cu in.
8	½	A	19.2	135.7	6.1	19.0	0.531	6.6	0.281
7	½	D	21.3	151.7	5.4	18.0	0.582	6.4	0.303
9	½	B	19.7	136.6	6.3	19.7	0.554	7.0	0.300
10	½	C	21.2	150.4	6.2	18.3	0.554	6.6	0.310
6	¾	D	21.9	237.6	4.8	15.9	0.538	6.1	0.302
1	1	A	19.4	278.8	5.8	18.0	0.516	7.2	0.312
2	1	A	19.1	272.2	6.1	18.6	0.516	7.3	0.311
5	1	D	21.9	318.2	4.9	16.0	0.536	6.5	0.323
4	1½	A	17.8	376.0	6.4	19.7	0.502	8.6	0.337
3	2	A	16.6	473.9	5.4	19.2	0.500	9.1	0.337

These results indicate that the conductivities calculated from the thermocouples generally employed are entirely consistent with the conditions that exist inside the material.

Moisture

It is well known that the moisture originally present, or acquired by absorption, plays a very important part in the heat transmission process, more particularly when, as in the case of refrigeration, it becomes frozen in use. The percentage of moisture present in the specimen is seldom quoted when conductivities are published or tabulated in books of reference. Table 7 gives the percentages of moisture present in the tests described which were obtained by heating at 140 F for 24 hours and are expressed as percentages of the dried weight.

Tests with varying quantities of moisture are difficult to make because some of the moisture travels from the hot to the cold side and its distribution cannot be controlled. To illustrate this point, two specimens of fibre board, 3½ in. thick and 24 in. square, were made up from boards that were 2 in. and 1½ in. thick, respectively. The thicker boards (2 in.) were placed next to the hot plate, and the 1½ in. boards next to the cold plate, and the test was conducted in the usual manner for 92½ hours. The mean temperature of the hot side was 101.8 F and that on the cold side was 50 F. During the test, the thin boards gained 0.188 and 0.186 lb in weight, respectively, the total increase being 0.374 lb. The thicker boards lost 0.141 and 0.140 lb, respectively, a total loss of 0.281 lb. Thus, the net result was a gain of 0.093 lb, but the boards nearest the cold plate gained in weight, and those nearest the hot plate lost weight. This indicates clearly the migration of moisture from the hot to the cold side. The calculated density for the entire specimen remained practically constant (15.6 lb per cubic foot) as the total difference in weight did not affect this appreciably.

However, there is no doubt that conductivity does increase with moisture content and Table 8 indicates the percentage increase in weight of samples of various insulating materials when exposed to a temperature of 70 F and a relative humidity of 85 to 90 per cent for 72 hours. The samples were protected from falling drops of water by a roof at about 18 in. above them and were exposed to air on all sides.

This indicates that, even when exposed to nearly saturated air for long periods, rock wool and glass wool absorb practically no moisture, and cork board absorbs very little.

The increase in weight of fibre board, however, called for further study of the amount of moisture absorbed by specimens of different thicknesses when exposed to air that was nearly saturated. Samples 8 in. square and of thicknesses varying from ½ in. to 2 in. were exposed to air at 65 F and 85 to 90 per cent relative humidity for 22 days, weights being obtained daily so that the progress of absorption could be noted. It was found that the samples were very sensitive to slight changes in the moisture content of the air, but finally steady values were obtained.

Exposure periods varying from 10 to 14 days were required for the different thicknesses to reach their maxima. The samples were then air dried at 70 F and 35 per cent relative humidity, and in 24 hours the ½ in. and ¾ in. specimens had lost practically all of the additional water acquired in the moist room. The 1 in. specimen required over 70 hours, but the 1½ in. and 2 in. specimens still retained 2 and 3 per cent of moisture (by weight), respectively, after 97 hours of air drying at 70 F and 30 per cent relative humidity. The samples were then dried in an oven at 140 F for 38 hours and were again weighed, this being taken as the dry weight upon which the percentages given in Table 9 are calculated. The various increases in weight are there expressed as percentages of the dry weight and in grammes per cubic inch of material.

It was considered, however, that the rates of moisture absorption through the cut surfaces might be different from those of the top and bottom faces. The tests were therefore repeated with specimens which were designed to have the same ratio of edge to total surface for each of the thicknesses represented. Duplicate specimens were made and were exposed to air at 70 F with 65 per cent relative humidity, and at 65 F with 85 to 90 per cent relative humidity, respectively. The results obtained are given in Tables 10 and 11 and are also illustrated by Fig. 11, which indicates the time taken to reach equilibrium in each case and the final state. It is evident that, with high humidities and the same ratio of

Table 10—Moisture Absorbed by Fibre Board Specimens

Geometrically Similar Specimens of Various Thicknesses Exposed to Air at 70 F and 65 Per Cent Relative Humidity

Sample No.	Dimensions In.	Thickness In.	Make	Density Lb/cu ft as Received	Dry Weight Grammes	Moisture as Received Per Cent	Final Moisture	
							Per Cent	Grammes per cu in.
4	3 x 3	½	A	19.2	18.0	7.2	8.9	0.356
5	3 x 3	½	A	19.2	18.4	7.1	8.7	0.356
6	3 x 3	½	D	21.3	21.65	5.8	8.5	0.410
7	3 x 3	½	D	21.3	21.55*	6.3	8.6	0.410
8	4½ x 4½	¾	D	21.9	74.45	5.1	7.3	0.358
9	4½ x 4½	¾	D	21.9	74.5	5.1	7.2	0.355
10	6 x 6	1	D	21.9	175.2	5.4	7.7	0.375
11	6 x 6	1	D	21.9	176.05	5.5	7.6	0.374
12	6 x 6	1	A	19.3	154.3	6.5	8.7	0.372
13	6 x 6	1	A	19.3	156.55	6.6	8.7	0.380
14	9 x 9	1½	A	17.8	475.4	6.4	9.5	0.372
16	12 x12	2	A	16.6	1065.5	5.9	9.1	0.338

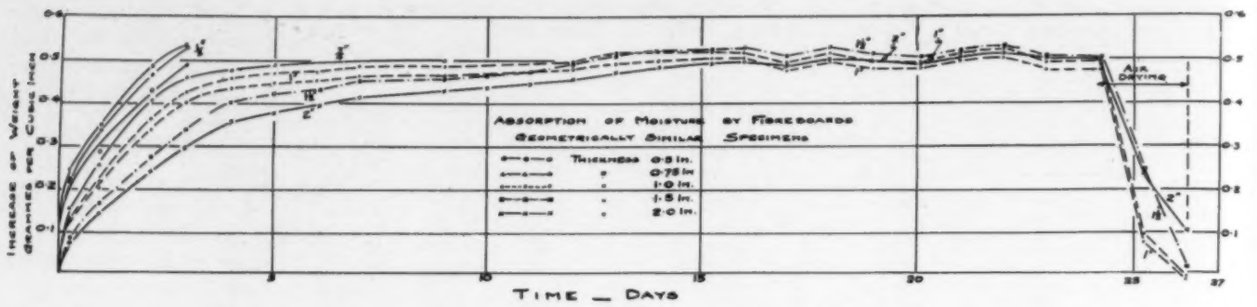

Fig. 11—Absorption of moisture by fibreboards

edge to surface, the weight of moisture absorbed is a function of the volume, as all of the curves reach a final absorption of about 0.5 grammes of water per cubic inch of volume starting from the weight *as received*. This is added to the original moisture content of about 0.23 grammes per cubic inch, giving a total moisture content of about 0.73 grammes per cubic inch. On a weight basis, this represents a total moisture absorption of 15 to 19 per cent added to the dry weight.

As applied to building construction, it is possible that these weights may be exceeded, if and when condensation occurs due to the cooling of heated air below the dew point, in which cases the air spaces may have a tendency to accumulate moisture. Insulating materials should absorb as little moisture as possible, and should also be treated or installed so that air infiltration is avoided altogether, or reduced to a minimum.

Conclusions

The experiments described indicate that valid comparisons cannot be made between heat transmission tests made on specimens of different sizes and thicknesses, and, in addition to the usual tabulation of mean temperatures and density, it is suggested that the moisture content should be stated. The following suggestions are made:

1. Two sizes only of hot plate should be standardized for use with thin and thick samples, respectively. Suitable dimensions would probably be 12 x 12 in. for specimens up to 1½-in. thick, and 24 x 24 in. for specimens up to 4- or 5-in. thick. The small plates are more convenient and give results in a shorter time than do large ones. The authors are in favour of replacing the guard ring by an adequate thickness of edge insulation. This would simplify the construction and cheapen the cost of the apparatus and would eliminate many difficulties of manipulation that are introduced by the guard ring. Also, if the latter is not correctly balanced it may introduce errors greater in extent than those it is designed to prevent. The authors have found by experience that edge losses can be reduced to negligible proportions by suitable insulation.

2. Mean temperatures should be standardized, preferably between 70 and 80 F. This question is directly related to that of the temperature difference between the hot and cold plates. Assuming that the latter is about 40 F, which appears to be a common value, the cold plate would be kept at 50 to 60 F, and the hot plate between 90 and 100 F. These temperatures avoid the undesirable formation of ice on the cold plates, and provide reasonable temperatures on the hot side. The advantage of this range of mean temperature is that it corresponds fairly well with the general range of atmospheric temperatures and tends to keep edge losses down to minimum values.

3. The percentage of moisture present in the specimen after testing should be measured, preferably by drying at about 140 F for 24 hours. This temperature was chosen because some insulating materials undergo structural changes when heated beyond that point (e.g. gypsum products). It is also desirable to record any change in weight that occurs during the test, and also the atmospheric temperature and relative humidity.

4. When publishing test results, the size, thickness and density of the specimen should be stated in inches and pounds per cubic foot, respectively.

5. The term *conductivity* should be replaced by some other word or phrase when this property is calculated, as is usual, from the conductance of the specimen. The phrase *apparent conductivity* has been suggested, but it may be advisable to avoid the implication of approximation which may be conveyed to some people by the word *apparent*.

Table 11—Moisture Absorbed by Fibre Board Specimens

Geometrically Similar Specimens of Various Thicknesses Exposed to Air at 65 F and 85-90 Per Cent Relative Humidity

Sample No.	Dimensions In.	Thickness In.	Make	Density Lb per Cu Ft as Received	Dry Weight Grammes	Moisture as Received Per Cent	Final Moisture		Moisture After Air Drying for 48 Hours	
							Per Cent	Grammes per Cu In.	Per Cent	Grammes Per Cu In.
1	3 x 3	½	B	19.7	18.35	5.4	18.8	0.766	5.4	0.222
2	3 x 3	½	B	19.7	18.25	5.8	18.9	0.766	5.8	0.233
3	3 x 3	½	B	19.7	18.50	5.9	18.9	0.778	5.4	0.222
4	3 x 3	½	A	19.2	18.60	5.9	17.7	0.723	5.1	0.211
5	3 x 3	½	A	19.2	18.30	5.7	17.7	0.721	5.4	0.222
6	3 x 3	½	D	21.3	21.40	5.6	16.8	0.800	5.2	0.244
7	3 x 3	½	D	21.3	21.25	5.6	16.7	0.790	5.4	0.255
8	4½x 4½	¾	D	21.9	72.35	4.5	15.0	0.715	4.8	0.227
9	4½x 4½	¾	D	21.9	74.75	4.5	14.8	0.728	4.7	0.233
10	6 x 6	1	D	21.9	177.3	4.9	15.0	0.743	4.8	0.236
11	6 x 6	1	D	21.9	172.0	4.8	15.0	0.718	4.6	0.222
12	6 x 6	1	A	19.3	154.0	6.1	17.2	0.736	5.3	0.225
13	6 x 6	1	A	19.3	150.85	6.1	17.3	0.725	5.2	0.216
14	9 x 9	1½	A	17.8	479.15	6.2	18.9	0.747	6.8	0.270
15	9 x 9	1½	A	17.8	478.6	6.1	18.9	0.744	6.7	0.264
16	9 x 9	2	A	16.6	1064.6	5.7	19.1	0.708	8.5	0.315

Acknowledgments

The thanks of the authors are due to the authorities of the University of Toronto for providing the necessary facilities for these experiments and also to C. Tasker of the Ontario Research Foundation who made the observations detailed in Table 10.

The Inspiration of Society Membership

When the A.S.H.V.E. was 20 years old, Samuel R. Lewis, consulting engineer, Chicago, Ill., was president and now after 20 odd years he has written some of his thoughts on Society membership in response to an invitation from R. C. Bolsinger, chairman of the Membership Committee. The comments of some A.S.H.V.E. members, who have been on the rolls for 25, 30, and 40 years are to be presented in this series of messages on such topics, as Why I Joined the Society; Why I Am Interested in the Society; and What I Value in Society Membership.

S. R. Lewis

WHY I JOINED THE SOCIETY

Because I heard that it was having a summer meeting in Chicago in 1904 and thought it worth while to drop in at one of the sessions held in the old Auditorium Hotel. Professor Kinealy was in the chair. I liked the way the members and guests helped each other by telling their experiences. I liked the genial manner of Professor Kinealy. I was a stranger and they took me in.

Such men as Thos. J. Waters, Chief Engineer of the Chicago Board of Education, August Kehm, a sterling business man as well as a sound engineer, E. F. Capron, who later became my valued partner, James Mackay, who helped me over many a knotty problem, and many others became my lifelong friends. Without the Society I would never have made such contacts.

We inaugurated the Illinois Chapter as a result of the social pleasure and professional benefit which were derived from the closer acquaintance engendered in the Society.

WHY I AM INTERESTED IN THE SOCIETY

Because of the Research Laboratory and The Guide. Both are unique, both were pioneering ventures, both are entirely without financial interest to any particular person or corporation. The quality of Wm. H. Driscoll always will be associated in my mind with the Laboratory, just as The Guide means Perry West, its outstanding originator.

WHAT I VALUE IN SOCIETY MEMBERSHIP

Contact with the leaders in the development of a great industry and some chance to participate with them in working out improvements.

The unique personalities of the members of the Society. They are universally lovable men, never self-seeking, always ready to serve and to give. Either the Society exerts a chastening influence on ordinary human nature, or the Committee on Admissions has been inspired.

WHY I AM GRATEFUL TO THE SOCIETY

Because even though the records will show that the ambitious self-seeker does not receive office or honor in the Society, I find that through the years my friends in the Society have diverted to my personal benefit many a profitable engineering venture. Why not acknowledge a real debt? Why not express a desire and a promise to reciprocate? Is not that after all the real pleasure in living?

Fifth International Heating and Ventilating Exposition

New York City has been chosen as the location of the Fifth International Heating and Ventilating Exposition to be held at Grand Central Palace, January 24-28, 1938, under the auspices of the A.S.H.V.E., when the 44th Annual Meeting of the Society will take place during the same week.

Since its inception in 1930, the Heating and Ventilating Exposition has been held biennially, successively in Philadelphia, Cleveland, New York, and Chicago. The presentation of the Exposition in leading cities in different parts of the country, together with the growing interest in air conditioning, has made it a most significant industrial exposition which has drawn attendance from all over the world. Each successive Exposition has met with an increasingly active response, both on the part of exhibitors and audience, and significant in connection with the Fifth Exposition is the trend to new building activity.

Three floors of Grand Central Palace have been reserved for the Exposition and invitations to participate have been extended to a number of organizations in the fields concerned with the heating, ventilating, and air conditioning industry. The conduct of the Exposition, including details of exhibit arrangement and leasing, will be in charge of the International Exposition Co., Grand Central Palace, New York, N. Y., under the personal direction of Charles F. Roth, manager, who has been similarly responsible for the earlier Expositions.

Refrigerating Machinery and Air Conditioning Manufacturers to Meet

Meetings of the *Refrigerating Machinery Association* and the *Air Conditioning Manufacturers' Association* have recently been announced by the respective presidents, J. M. Fernald and J. F. G. Miller.

The *Refrigerating Machinery Association* will meet on May 12 and 13 and the *Air Conditioning Manufacturers' Association*, will hold their meeting on May 14 and 15, in the Homestead, Hot Springs, Va.

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS *is located*

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Committee on Radiation with Gravity Air Circulation—IP-9: M. K. Fahnestock, *Chairman;* H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson,* R. N. Trane, T. A. Novotney.

Committee on Relation of Body Changes to Air Changes—OH-3: Dr. E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, John Howatt, A. P. Kratz,* P. J. Marschall, V. L. Sherman.

Committee on Sound Control—IF-1: J. S. Parkinson, *Chairman;* C. M. Ashley, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, A. E. Stacey,* F. R. Watson.

Committee on Summer Air Conditioning for Residences—IP-7: M. K. Fahnestock, *Chairman;* E. A. Brandt, John Everetts, Jr., Elliott Harrington,* H. F. Hutzel, E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick, J. H. Walker.*

Committee on Treatment of Air with Electricity—C-17: Prof. C.-E. A. Winslow,* *Chairman;* R. D. Bennett, W. H. Carrier, L. W. Chubb, Major W. D. Fleming, R. F. James, L. R. Koller, Dr. C. A. Mills, Prof. E. B. Phelps, Prof. G. R. Wait, Prof. W. T. Wells.

*Member of Committee on Research.

A. S. H. V. E. RESEARCH

A Page of News Notes and Comment

By **Brewster S. Beach***

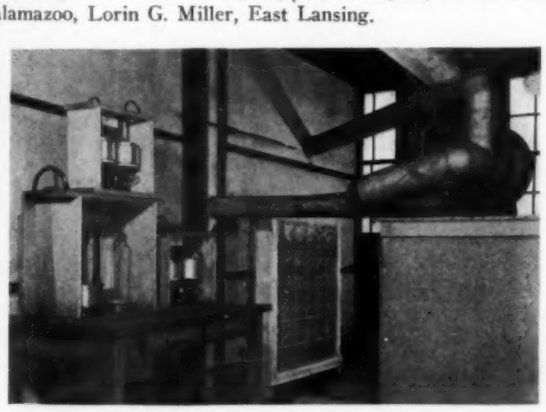
NEW RESEARCH REPRESENTATIVE PLAN

A plan to achieve closer contact between A.S.H.V.E. local chapters and the Committee on Research of the Society, by having Research Representatives appointed by each chapter, was announced in April by W. A. Danielson, chairman of the committee. "This closer contact, and resulting influence of the chapters," Col. Danielson said, "will prove, I am convinced, a powerful factor in helping the committee achieve its objectives of broad research in the interests of the Society."

Each research representative under this plan will act as a liason officer between the members of his chapter and the committee's chairman. Among other duties it is expected that Research Representatives will reflect ideas for research activity as gathered from the chapter memberships and function as a local contact with the press for the dissemination of news about the Society's research program.

As the plan takes effect, Col. Danielson expects it to result in valuable suggestions for new research or the direction of existing work along channels which would secure the maximum interest and support from the local chapters and from sources upon which the Committee on Research depends for a portion of the funds required to finance research.

As this news page went to press eight chapters had appointed Research Representatives as follows: Pittsburgh Chapter: F. C. McIntosh; Western New York Chapter, Buffalo, New York: David J. Mahoney; Ontario Chapter, Toronto: Ernest R. Gauley; Montreal Chapter: F. J. Friedman; Minnesota Chapter, Minneapolis: Milton Wunderlich; Michigan Chapter, Detroit: Hugo F. Hutzel; Winnipeg Chapter: George C. Davis; Western Michigan Chapter: W. W. Bradfield, Grand Rapids, S. H. Downs, Kalamazoo, Lorin G. Miller, East Lansing.



Test set-up for glass building units at Pittsburgh Testing Laboratories, supervised by M. L. Carr, chairman, A.S.H.V.E. Committee on Glass Requirements.

FIELD TEST ON SUMMER COOLING

To test the validity of the recent work on Requirements for Summer Comfort Cooling, the A.S.H.V.E. Laboratory is completing details of a project to study the comfort reactions of a large group of office workers employed by the Minneapolis-Honeywell Regulator Company who have agreed to turn over their air conditioned general offices for this investigation.

F. C. Houghten went to Minneapolis recently to decide on final arrangements. The investigation will involve some 200 people, representing a typical cross-section of persons who frequent stores, theaters and other types of commercial establishments. Arrangements under which it will be possible to change the temperature, humidity and air movement within the spaces occupied by these workers will be made.

This study is part of the research program into comfort cooling which began two years ago and will be continued again this summer at Pittsburgh, Texas A. & M. College, and at the Ontario Research Foundation, Toronto. The chairman of the technical advisory committee in charge of this work is C. Tasker and consists of A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington, Dr. E. V. Hill, R. E. Keyes and C. P. Yaglou.

The coordination of the research work, which was started last year by means of group chairmen, is being continued this year. There are four groups of committees—designated as follows:

Organic—Human—(OH)
Inorganic—Fundamental—(IF)
Inorganic—Performance—(IP)
Chairman—(C)

The (C) Group comprises those which pertain to all phases, or for other reasons can be best handled directly by the Chairman.

The Group Chairmen carry out the greater part of the duties of the Chairman as prescribed by the regulations governing the Committee on Research.

The 1937 Chairman of the OH Group is Walter L. Fleisher, and the chairman of the IP Group is Thomas H. Urdahl. A chairman for the IF Group will be announced shortly.

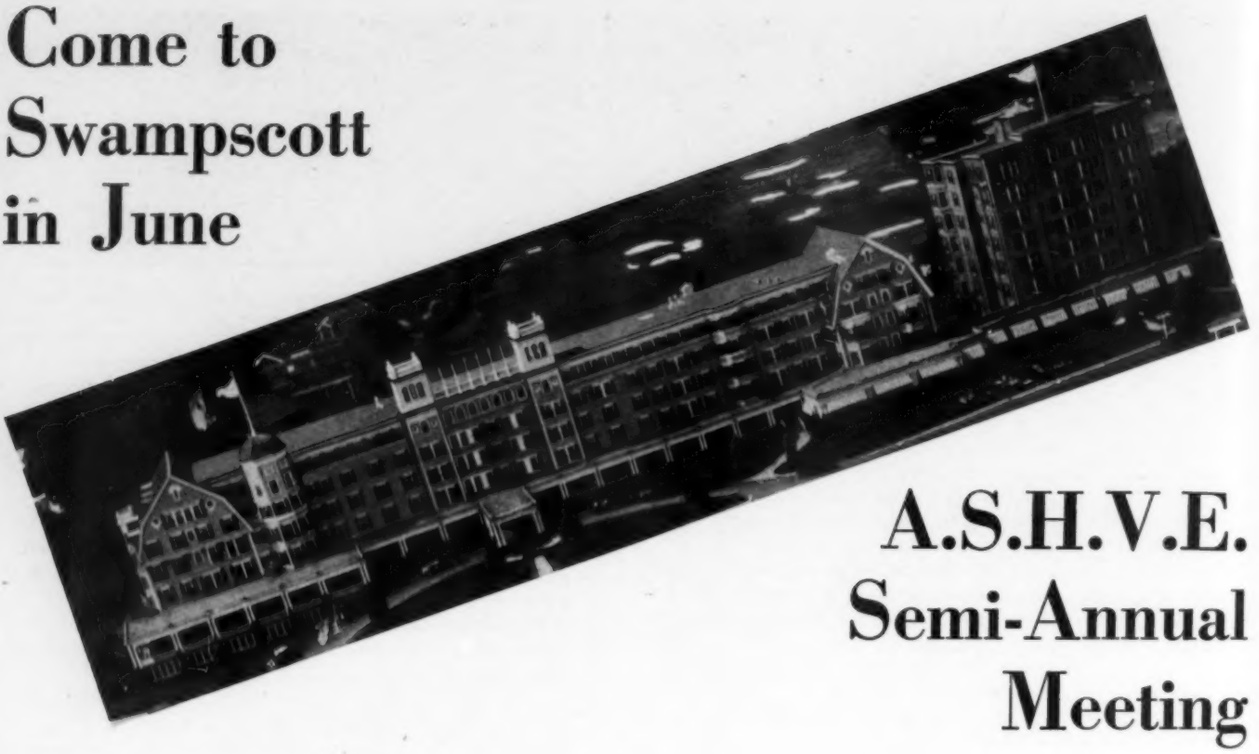
W. A. Danielson, *Chairman*
COMMITTEE ON RESEARCH

Roofs take to the water to keep building cool in summer, warm in winter!

Water insulating pool atop Swann & Company's two story air conditioned laboratory at Birmingham, Alabama. The Research Committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is planning an investigation of this subject to collect data for the air conditioning industry.

*Director, Information Service, Committee on Research.

Come to Swampscott in June



A.S.H.V.E. Semi-Annual Meeting

New Ocean House . . . . . June 24-26, 1937

An Invitation

The Massachusetts Chapter sends its greetings and welcomes you to take part in the Summer Meeting of your Society at Swampscott on June 24, 25, 26, 1937.

Swampscott on the shore of the Atlantic Ocean lies near the center of a great industrial, historical and marine area of the old Bay State.

A program that you will want to participate in has been prepared—to bring to you the latest developments in our industry, combined with much enjoyable entertainment among the cooling ocean breezes.

A host of special arrangements has been made for the ladies which they will particularly enjoy and long remember. At every step you will find innumerable points of interest in the "Cradle of Liberty." To many, the ocean with its many attractions, diversions and moods will be a new and thrilling experience.

Plan now to be with us at Swampscott. We want to meet you and greet you again.

JAMES HOLT

President, MASSACHUSETTS CHAPTER

Lobby of the New Ocean House

The Massachusetts Chapter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is preparing a hospitable welcome for members attending the Semi-Annual Meeting 1937 to be held at the New Ocean House, Swampscott, Mass., June 24-26, 1937.

The Meetings Committee has selected papers dealing with the latest developments in the industry which will be presented coupled with an extraordinary program of entertainment arranged by the Chapter.

For several months the Committee, of which W. T. Jones is Chairman, has been planning the details of the meeting. Council meetings, committee meetings, registration, luncheons and lectures will be held at the New Ocean House, which is on the beautiful North Shore of Massachusetts Bay just a few miles from Boston. Hotel rates are on the American Plan and range from $6.00 to $10.00 per day per person depending on type of accommodation and location.

The ocean nearby will be the delight of those who enjoy water sports, boating, and deep sea fishing. Special arrangements by the Ladies Committee already include trips to the renowned Harvard glass flowers, nearby estates, gardens and local historical sites. There will be trips to Marblehead and Gloucester for the members and ladies.

A golf course located at the hotel will attract many and those who enter the tournaments will enjoy Tedesco Country Club.

There is every incentive to come to Swampscott, June 24-26, for a worthwhile meeting and a few days of recreation. Plan the trip now—New England is delightful in June.

TENTATIVE PROGRAM

(Daylight Saving Time)

Wednesday, June 23, 1937

10:00 a. m. Committee Meetings
1:00 p. m. Registration
2:30 p. m. Council Meeting
8:30 p. m. President's Reception, Music and Dancing

Thursday, June 24, 1937

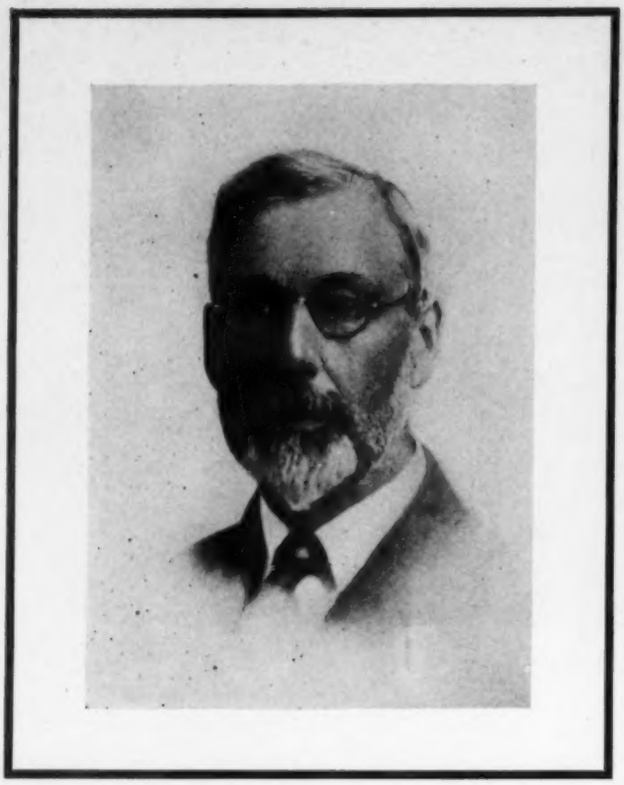
9:00 a. m. Registration
9:30 a. m. Technical Papers—
The Adaptability of Pre-Cooling Coils to Air Washer Systems, by John Everetts Jr.
Degree Days for Summer Air Conditioning, by A. D. Marston
Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 and 46 F, by A. P. Kratz, S. Konzo and E. L. Broderick
Cooling Load Analysis of a Bank Building, by John James
2:00 p. m. Golf Tournament—Tedesco Country Club
2:30 p. m. Boat Trip to Marblehead and Gloucester
8:30 p. m. Informal Party, Concert and Dancing

Friday, June 25, 1937

9:30 a. m. Technical Papers—
Ventilation Requirements (Part II), by C. P. Yaglou and W. N. Witheridge
Performance of Fin Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman
The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick
Report of A.S.H.V.E.-A.S.R.E. Committee on National Standards for Air Conditioning Applications, by L. A. Harding, *Chairman*
10:00 a. m. Bus leaves for Boston—Ladies to visit Harvard Museum and Glass Flower exhibit
2:00 p. m. Golf Tournament, Tedesco Country Club
2:30 p. m. Ladies Bridge and Tea—Tennis, Archery and other Sports
7:30 p. m. Banquet (New England Shore Dinner) and Dance —(Presentation of Past President's Memory Book)

Saturday, June 26, 1937

9:30 a. m. Technical Papers—
Simplified Method of Testing and Rating Air Cleaning Devices, by Arthur Nutting
Direct Reading Effective Temperature Indicator, by J. R. Parsons
Infiltration Effects on Heating Grant Building, by F. C. Houghten
10:00 a. m. Ladies visit to North Shore Gardens



Death of Champlain L. Riley

Champlain L. Riley, a member of the firm of Clark, MacMullen & Riley, Inc., consulting engineers, New York, N. Y., and president of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in 1921, died suddenly of a heart attack at his home on April 16, 1937.

Mr. Riley was born in East Orange, N. J., on March 2, 1877, the son of William H. and Louisa Lord Riley. His education was obtained in the high schools at Plainfield, N. J., and Ithaca, N. Y., where he also attended Cornell University, receiving the degree of M.E. in E.E. in June 1899.

Immediately after graduation Mr. Riley became associated with the firm of Clark & MacMullen, working successively as draftsman and superintendent, director and secretary. He supervised the design and superintended the installation of heating, ventilating and electric lighting equipment in many public and industrial buildings. Later Mr. Riley became a member of the firm of Clark, MacMullen & Riley, Inc., with offices in New York City and Cleveland, O. and of MacMullen, Riley & Angus, Toronto. Among the buildings for which they have been consultants for the mechanical and electrical engineering requirements are Severance Hall, Cleveland, Chanin Building, New York, American Salpa Corp., East Spotswood, N. J., Federal Reserve Bank, Cleveland, Crosse & Blackwell, Ltd., Baltimore, Pennsylvania Textile Mills, Clifton, N. J., and Riker's Island Penitentiary, New York.

Mr. Riley joined the A.S.H.V.E. in 1906, and he will be remembered for the many splendid services he rendered as president in 1921, as first vice-president in 1920, as a member of the Council from 1918 to 1922 and Advisory Council from 1924 to 1937. He was always alert to advance the interests of the engineering profession and served on many committees of the Society with distinction. Among the committees on which he served are the following: 1917—representative on Census Committee of Engineering Council; 1918—Publication Committee and as chairman in 1919, 1920 and 1922; 1921-1922—Executive Committee and as chairman in 1921; 1922—Committee on Industrial Reconstruction; 1922-1925—Committee on Revision of Constitution and as chairman in 1922 and 1923; 1922, 1923, 1926—Guide Publication Committee; 1923-1925—Committee to Confer with *American Institute of Architects;* 1923—chairman of Committee to Consider the Report of the New York State Commission on Ventilation; 1926-1927—Committee to Cooperate with Rochester School Board; 1931-1934—Committee on Ventilation Standards; and from 1924 as a member of the Advisory Council.

Mr. Riley was the twenty-eighth President of the Society and his administration was distinguished by several important events in the history of the organization. During his administration the late Dean F. Paul Anderson was appointed Director of the Society's Research Laboratory and in October 1921 THE A.S.H.V.E. GUIDE was announced. In Mr. Riley's address as President of the Society his words concerning THE GUIDE were prophetic, "THE GUIDE will prove a great help in financing the Society's many activities." He was greatly interested in this project and served on the first Guide Publication Committee, the personnel of which consisted of:—Perry West, *Chairman,* Homer Addams, Treasurer, J. Esten Bolling, Secretary, W. H. Driscoll, J. I. Lyle, S. E. Dibble, E. S. Hallett and C. L. Riley. The Engineers Code of Ethics was adopted while Mr. Riley was President, and a tablet in the memory of John R. Allen, first Director of the Research Laboratory, was erected in October 1921, at the U. S. Bureau of Mines.

Mr. Riley was active in civic and public affairs in Plainfield, N. J., where he resided. He was one of the organizers of the New Jersey Men's League for Woman's Suffrage and in recent years helped to found the public forum of the *First Unitarian Society.* He and Mrs. Riley were instrumental in obtaining Parker Field, Plainfield, now known as High School Field, as a recreation center for the public schools.

Surviving Mr. Riley are his widow, Mrs. Ida Holt Riley, three sons, Philetus, Edward and Richard, a daughter, Mrs. Constance Riley Richards, and a brother, Howard Wait Riley.

In the passing of Mr. Riley the profession loses a distinguished member and a loyal friend. The Society regrets the untimely death of an outstanding member who has served it well and the Officers and Council extend their sincerest sympathy to his family whose loss is irreparable.

Symbols Group Meets

A meeting of the sub-group of the *American Standards Association* on Graphical Symbols for Heating, Ventilating, Refrigeration and Air Conditioning was held in New York City on April 19. It is the purpose of this group to study existing symbols for use in drawings now available in the field and expand the previously adopted list to include equipment and apparatus which is now being generally used in this field of application.

The chairman of the committee is E. E. Ashley, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, with the following additional members: L. L. Munier and S. Lewis Land, *Heating, Piping and Air Conditioning Contractors National Association;* J. L. Murphy, *National Association of Master Plumbers of the United States;* E. L. Penfrase, *American Society of Sanitary Engineering;* L. H. Roller, *American Society of Refrigerating Engineers;* and E. R. Wolfert, *Air Conditioning Manufacturers Association.*

A.S.R.E. to Meet at French Lick

Announcement has been made by D. L. Fiske, secretary of *American Society of Refrigerating Engineers,* that the spring meeting of the Society will be held at French Lick Springs Hotel, French Lick, Ind., June 8-10.

NEWS OF LOCAL CHAPTERS

OFFICERS OF LOCAL CHAPTERS—1937

ATLANTA: *Organized,* 1937. *Headquarters,* Atlanta, Ga. *President,* E. W. KLEIN, 152 Nassau St., N. W. *Secretary,* C. T. BAKER, 713 Glenn St., S. W.

CINCINNATI: *Organized,* 1932. *Headquarters,* Cincinnati, O. *Meets,* Second Tuesday in Month. *President,* E. B. ROYER, 6635 Iris Ave. *Secretary,* O. W. MOTZ, 920 E. McMillan St.

ILLINOIS: *Organized* 1906. *Headquarters,* Chicago, Ill. *Meets,* Second Monday. *President,* J. J. HAYES, 53 W. Jackson Blvd. *Secretary,* C. E. PRICE, 6 N. Michigan Ave.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* L. R. CHASE, 2440 Pennway. *Secretary,* F. J. DEAN, JR., 6028 Walnut St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg, Man. *Meets,* Fourth Thursday. *President,* J. B. STEELE, 184 Waterloo St. *Secretary,* C. H. TURLAND, 325 Centennial St.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Second Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* JOHN TURNER, 285 Columbus Ave.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich. *Meets,* First Monday after the 10th of the Month. *President,* R. K. MILWARD, 127 Campbell Ave. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* L. G. MILLER, Mich. State College, E. Lansing, Mich. *Secretary,* E. I. ADAMS, 115 So. Pine, Lansing, Mich.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* N. D. ADAMS, 220 Second Ave. S. W., Rochester, Minn. *Secretary,* J. E. SWENSON, 800 Hennepin Ave.

MONTREAL: *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* G. E. OLSEN, 101 Park Ave. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* P. S. HEDLEY, Curtiss Bldg. *Secretary,* C. A. GIFFORD, 758 Parkside Ave.

NORTHERN OHIO: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* L. T. AVERY, 2341 Carnegie Ave. *Secretary,* W. R. BEACH, 75 Public Sq.

OKLAHOMA CITY: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday. President, F. X. LOEFFLER, 1604 N. W. Fifth St. *Secretary,* E. W. GRAY, Box 1498.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* THOMAS McDONALD, 117 Peter St. *Secretary,* H. R. ROTH, 57 Bloor St. W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* S. D. PETERSON, 473 Colman Bldg. *Secretary,* D. C. GRIFFIN, Orpheum Bldg.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* L. P. HYNES, 240 Cherry St. *Secretary,* C. B. EASTMAN, 530 Brookview Lane, Brookline, Upper Darby, Pa.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* M. L. CARR, P. O. Box 1646. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* C. R. DAVIS, 2328 Locust St. *Secretary,* R. J. TENKONOHY, 1474 S. Vandeventer.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* LEO HUNGERFORD, 1057 N. La Brea Ave. *Secretary,* H. M. HENDRICKSON, 5051 Santa Fe Ave.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* F. E. GIESECKE, Texas A. & M. College. *Secretary,* W. H. BADGETT, Texas A. & M. College.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* W. A. DANIELSON, 2236 Munitions Bldg. *Secretary,* M. D. KICZALES, 3000 Connecticut Ave., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* M. E. ERICKSON, 8405 W. Greenfield Ave., West Allis, Wis. *Secretary,* R. G. KOCH, 626 E. Wisconsin Ave.

Atlanta Members Interested in Organizing Chapter

On March 25 a conference and luncheon was held at the Atlanta Biltmore Hotel, Atlanta, Ga., and was attended by 29 members of the Society and others interested in the possibilities of organizing a local chapter in that city.

The meeting was called to order by C. T. Baker who explained that several members had made inquiry about organizing a local chapter and he had invited the group to attend a luncheon to learn the procedure for taking such action from the Secretary of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, A. V. Hutchinson.

It was voted to have Mr. Baker act as Chairman and with H. King McCain investigate the desire of other members for presenting a petition for a chapter charter to the Council of the Society.

It was decided to hold a meeting on April 22 at which time reports on the organization work were scheduled for presentation.

Those in attendance were: C. T. Baker, K. E. Gouedy, R. F. Hahn, H. King McCain, A. V. Hutchinson, J. S. Adams, L. L. Barnes, S. W. Boyd, R. C. Broach, C. E. Brockinton, Boynton Cole, G. T. Gunnell, R. D. Kelly, E. P. Kramer, I. W. Liggett, A. L. Lindstrom, C. H. McLeod, W. R. Moore, R. B. Stotz, T. M. Turner, M. W. Wise, R. R. Yarborough, R. O. Bennington, C. C. Gray, C. P. Goree, Jr., M. G. Keiser, J. Kilian, E. W. O'Brien, E. L. Shuff.

Birmingham Group Holds Meeting

On March 23 a luncheon meeting was held at the Bankhead Hotel, Birmingham, Ala., when local A.S.H.V.E. members and others interested in heating, ventilating and air conditioning had the opportunity of hearing Secy. A. V. Hutchinson discuss Society and chapter organization and activities.

Interest in holding informal meetings has been stimulated and those who attended the luncheon were: C. P. Lichty, H. V. Fried, R. L. Murphree, A. O. Festorazzi, F. L Hardy, Haydn Myer, H. K. Walden, M. D. McRae, Jr., F. A. Witt, L. E. Hoene, Barney DeRamus, M. B. Hinman, C. R. Crowder, C. E. Beck, V. A. Smith, J. T. Badham, E. J. Tully, W. A. Finley, E. M. Shelton, J. S. Fox, Milton Tyler, Jr., H. E. Newman, R. A. Sansing, Jr., George Stafford, J. G. Trawick, J. L. LaPointe, A. J. Lichty, L. T. Cale, and Hugh Roberts of Birmingham, R. F. Connell, Detroit, and A. V. Hutchinson, New York

Use of Fiber Glass, Air Filters and Glass Block Presented to Ontario Chapter; E. H. Gurney Speaks on Round the World Trip

April 2, 1937. One hundred and sixty-five members and guests attended the meeting of Ontario Chapter at the Royal York Hotel, Toronto. Following dinner, Secy. H. R. Roth read the report of the Nominating Committee.

Pres. Thomas McDonald announced a joint luncheon of engineering societies to be held at the Royal York Hotel, April 13, to which the A. S. H. V. E. members were cordially invited, and on this occasion Prof. R. W. Angus was to be the principal speaker.

A report on the Code Committee was given by M. W. Shears.

Upon motion properly seconded, it was voted that E. R. Gauley be the Ontario Chapter's Research Publicity Representative.

D. O. Price introduced the principal speakers of the evening, F. Meyers and E. P. Lockart, who spoke respectively on the Use of Fiber Glass and Air Filters, and Glass Block in Construction.

March 1, 1937. The regular March meeting of Ontario Chapter was attended by 53 members and guests.

Pres. Thomas McDonald nominated C. Tasker as chairman of a committee to investigate the possibility of having Dr. C. A. Mills, Cincinnati, O., as the principal speaker at one of the fall Chapter meetings.

It was moved by E. J. McDonald and seconded by E. H. Gurney that the Board of Governors appoint a committee for formulating codes. This committee is to appoint sub-committees for various codes, covering, for example, boiler, warm air furnace, and insulation installations.

Mr. Gurney, first vice-president of the A. S. H. V. E., was the principal speaker of the evening and was introduced by H. B. Jenney. Mr. Gurney, who recently returned to Toronto from a trip around the world, spoke on Heating, Ventilating and Air Conditioning in South Africa.

Secy. H. R. Roth reports that H. H. Angus extended a vote of thanks to Mr. Gurney on behalf of the Chapter for his splendid address, and the meeting then adjourned.

Refrigeration and Air Conditioning Discussed at Western New York Meeting

March 8, 1937. Pres. P. S. Hedley called the meeting of Western New York Chapter to order at the University Club with 49 members and guests present. The minutes of the last meeting were read and approved.

W. E. Voisinet gave a report which was favorable to cooperation by the Chapter with the Buffalo Sheet Metal Warm Air Heating and Air Conditioning Association.

After several announcements, President Hedley presented the speakers of the evening.

Joseph Davis, assisted by Joseph Scanlon, discussed the subject of Application of Refrigeration and Air Conditioning. Mr. Davis traced the start of refrigeration and air conditioning, and Mr. Scanlon further developed the discussion of the subject with lantern slides. After the conclusion of the talk, many interesting questions were answered.

Samuel Sklar, a former chapter member who had just returned from a trip to the Orient, told of his experiences and spoke of the advantages of being an American.

At the suggestion of President Hedley, the speakers were accorded a rising vote of thanks.

Michigan Chapter Meets at University of Detroit

March 15, 1937. After a dinner at Corby's Inn about 110 members and guests met at the Physics Laboratory of the University of Detroit. Pres. R. K. Milward called the meeting to order and introduced Dean C. J. Freund, who welcomed the Michigan Chapter.

Prof. F. J. Linsenmeyer was presented and gave a talk on some of the physical phenomena of air conditioning, including hygrometry, psychrometry, and dew point. Insulation was also discussed and the talk was illustrated by slides.

Following Professor Linsenmeyer's address, Professor Aberle gave a demonstration of the various instruments used in air conditioning and conducted experiments illustrating their use.

The meeting adjourned at 11:00 p. m,. according to the report of Secy. G. H. Tuttle.

February 8, 1937. The Michigan Chapter of the A. S. H. V. E. held their annual Valentine dinner dance at Dearborn Inn and the 112 present included members, guests and friends.

Music was furnished by David Diamond and his orchestra until the party ended at 1:00 o'clock. According to Secy. G. H. Tuttle everybody had a very delightful evening.

A. W. Moulder Is Speaker at Montreal Chapter Meeting

March 15, 1937. Thirty-eight members and guests of the Montreal Chapter attended the meeting held at the Windsor Hotel, and following dinner Pres. G. L. Wiggs called the assembly to order. The roll was taken and the guests were introduced to the members.

A. W. Moulder, vice president in charge of engineering, Grinnell Co., Providence, R. I., was the guest speaker of the evening, and was introduced by President Wiggs. Mr. Moulder gave an illustrated lecture on Forced Hot Water Heating, and discussed the economical advantages of hot water heating due to its adaptability to regulation, off-setting outside weather conditions, and its ability to meet structural conditions; low maintenance cost, due to the absence of working parts; low depreciation and first cost.

The speaker then discussed in detail heat regulation afforded by orifices, baffle plates, etc., giving a history of the development of orifices and also engineering information on design. Mr. Moulder felt that there was a considerable future for hot water heating as it lends itself as an aid to summer cooling.

After the lecture a vote of thanks was extended on behalf of the Chapter to the speaker, by G. L. Ballantyne.

The meeting adjourned at 9:40 p. m., according to the report of Secy. C. W. Johnson.

Air Conditioning by Ionization Subject of Western Michigan Meeting

March 18, 1937. Western Michigan Chapter held its regular meeting in the English Room of the Rowe Hotel, Grand Rapids, and the roll call showed that 20 members and 40 guests were present.

In the absence of Pres. L. G. Miller, Dean H. B. Dirks, of Michigan State College, presided, and introduced the speaker of the evening, G. W. Penney, Westinghouse Electric & Manufacturing Co., who gave a very interesting talk on Air Conditioning by Ionization with suitable illustrations and demonstrations. The talk was enthusiastically received and was discussed by the entire group present, according to the report of Secy. E. I. Adams.

Characteristics of Glass Block Discussed at Cleveland Meeting

March 15, 1937. Approximately 50 members and guests attended the meeting of Cleveland Chapter, which was preceded by a dinner at which 25 were present.

The minutes of the past two meetings were read and accepted.

Pres. L. T. Avery announced that the Chapter would be honored by the presence of E. H. Gurney, Toronto, 1st vice-president of the Society, at the April meeting.

E. W. Gray reported for the Membership Committee and F. A. Kitchen reported for the Legislation Committee. The report of the treasurer was given by H. M. Nobis.

A resolution was moved by C. F. Eveleth and seconded by Philip Cohen that Secy. W. R. Beach be instructed to draw up a letter opposing Ohio State Senate Bill No. 83, the letter to be approved by the Board of Governors before being forwarded.

President Avery then turned the meeting over to Mr. Eveleth, who was acting chairman of the Program Committee. George Rigby was introduced and presented the speaker of the evening, E. P. Lockart, Owens-Illinois Glass Co., Toledo, O.

Mr. Lockart gave a brief history of glass and explained the characteristics of a newly developed glass block. His talk was accompanied by slides and by moving pictures.

Hot Water Heating Discussed at Washington, D. C., Meeting

March 10, 1937. Twenty-six members and guests were present for dinner at the Hotel Harrington, while 60 attended the meeting of Washington, D. C., Chapter which was called to order at 8:00 p. m.

Pres. W. A. Danielson presided and the minutes of the preceding meeting were read and approved as corrected.

E. V. Fineran was appointed by President Danielson to replace J. B. Hewett as a member of the Nominating Committee.

Announcement was made of the Chapter meeting on April 9, when members of the Council and officers of the Society planned to be present as guests of the Chapter.

A. W. Moulder, vice-president, Grinnell Co., Providence, R. I., was introduced by President Danielson and spoke on Forced Hot Water Systems. Mr. Moulder prefaced his talk with an explanation of the advantages of hot water heating and described various forced hot water heating installations, a number of which were in industrial buildings. With the aid of lantern slides, Mr. Moulder showed how many of the architects' objections to exposed piping, etc., were overcome by the use of copper-lined and concealed hot water fin type convectors.

C. E. Pullum, vice-president, Bell & Gossett Co., Chicago, Ill., was then introduced and spoke also on Hot Water Heating Systems. With the aid of lantern slides, Mr. Pullum explained a simple method used in designs of one-pipe and two-pipe reversed or direct return systems.

A lively discussion, entered into by the members and guests, followed the talks by Mr. Moulder and Mr. Pullum, who capably answered all questions within the scope of their experience.

Secy. M. D. Kiczales reports that after a rising vote of thanks to the speakers, the meeting adjourned at 11:00 p. m.

Air Conditioning Installation Using Well Water Described to Illinois Chapter

March 8, 1937. Illinois Chapter met at the Hotel Sherman when 135 were present including 13 guests from Wisconsin Chapter.

Pres. J. J. Hayes opened the meeting with a welcome to the Wisconsin guests, to which M. E. Erickson, president of the Chapter responded. Before the regular business of the evening was started, President Hayes called on Prof. G. L. Larson, Madison, Wis., past president of the A. S. H. V. E.

Minutes of the preceding meeting, as well as a report of the 43rd Annual Meeting of the Society in St. Louis, were read by Secy. C. E. Price. The recent death of Macy Good, a former member of the Chapter, was announced and the applications for membership of O. J. Prentice and W. R. Knudsen were read and accepted.

R. E. Hattis was the principal speaker of the evening and described an air conditioning installation designed for the windowless building of National Aluminate Corp. Well water was available for cooling, the equivalent of 90 tons of refrigeration being secured for 13 or 14 hp. Simultaneous heating and cooling were made possible by means of a heating coil and a cooling coil in each branch duct and a very complete control system. Mr. Hattis explained these methods in detail, as well as how condensation from humidity was prevented in winter, the handling of outside and recirculated air, and the cleaning of air. The discussion which followed brought out many interesting features.

Since the installation described by Mr. Hattis used well water for cooling, the subject of deep wells was an interesting accompaniment which was capably treated by H. P. Barton. It was pointed out that ground water supplies vary greatly in different localities and sometimes in just a few feet of distance. Mr. Barton told of the different water bearing strata, their depths, the gallons per minute to be expected from each and the temperatures usually obtained. It seems, however, that the ability to locate these strata and drive a well with the assurance that it will produce is still uncertain, according to the speaker. Mr. Barton also briefly discussed types of well pumps.

The results of the votes for members of the Chapter's Nominating Committee were announced as follows: John Howatt, C. E. Price, J. H. Milliken, J. J. Hayes, J. J. Aeberly, C. W. Johnson, and C. W. DeLand.

Los Angeles Holds Joint Meeting with Refrigerating Engineers

March 3, 1937. The Southern California Chapter of the A. S. H. V. E. held a joint meeting with the local *A. S. R. E.* Section on the occasion of the visit of D. L. Fiske, secretary of the *American Society of Refrigerating Engineers*. Dinner was served at 6:15 p. m. at the Clark Hotel, Los Angeles, with 108 members and guests present, who included by invitation local members of the founder engineering societies and the *N. A. P. R. E.*

Following dinner, Chairman J. C. Blair of the *A. S. R. E.* called the meeting to order and welcomed the members of the A. S. H. V. E., the *N. A. P. R. E.*, and other engineering societies. The program was opened with instrumental and vocal music by the Swiss Yodelers.

Mr. Nelson of the *N. A. P. R. E.* was called upon to announce their Jamboree at the Breakfast Club on March 25, and the Annual Pacific Coast Conference meeting at Portland, Ore., April 2 and 3.

Pres. Leo Hungerford spoke for the A. S. H. V. E. and announced a joint meeting with the *A. S. R. E.* during April coinciding with Pres. D. S. Boyden's visit to the Southern California Chapter. He also spoke of a joint meeting scheduled for May with Prof. B. M. Woods, chairman mechanical engineering department, University of California at Berkeley, as the principal speaker.

Two vocal numbers were given by R. E. Ristow, accompanied by W. L. Holladay at the piano.

Mr. Holladay was called upon to conduct the remainder of the meeting and he expressed the pleasure of the membership on having Mr. Fiske present. Mr. Holladay asked all of the past presidents of the local *A. S. R. E.* to stand so that he might introduce them, and he also presented President Schindler, of the Los Angeles Engineers Club.

Mr. Holladay then told the assemblage something of the work and experiences of Mr. Fiske, who gave a very interesting address on How Not to Run a Business.

Considerable discussion, led by Mr. Holladay, took place after Mr. Fiske's views on discipline and intelligence were expressed. Among those taking part were Messrs. F. G. Baender, M. I. Blumenthal, Herman Vetter, N. H. Rosberg, J. C. Blair, Knapp, Royden, Lincoln, and White.

According to the report of Secy. H. M. Hendrickson, an interesting time was had by all and the meeting adjourned at 10:00 p. m.

Philadelphia Chapter Elects Officers

April 8, 1937. At the regular meeting of Philadelphia Chapter Pres. W. F. Smith presided and called the assembly to order at 8:00 p. m.

Mr. MacIntosh spoke of the purpose of the United Campaign and asked for the cooperation of the Chapter membership in this work.

The minutes of the previous meeting and the report of the treasurer were read and approved.

The application of a new chapter member was reported by A. E. Kriebel, chairman of the Membership Committee.

R. C. Bolsinger, chairman of the Society's Membership Committee, asked for the cooperation of the Chapter in his efforts

to increase membership in the A. S. H. V. E.

L. E. Moody, chairman of the Auditing Committee, stated that the accounts of the treasurer were in proper order.

Secy. H. H. Erickson read the report of the Nominating Committee for Chapter officers as follows:

President—L. P. Hynes
Vice-President—H. H. Erickson
Secretary—C. B. Eastman
Treasurer—R. F. Hunger
Board of Governors—W. F. Smith, A. E. Kriebel, H. H. Mather.

It was duly moved that nominations be closed and the secretary was then instructed to cast a ballot for the above nominees.

After the election of officers, C. B. Eastman, chairman of the Meetings Committee, introduced Paul Young, of Johns-Manville Corp., who presented a paper entitled, Sound Insulation and Isolation. This was accompanied by a sound film, and proved to be interesting as well as instructive. The presentation presented the importance of sound or noise control in connection with air conditioning.

A. M. Simpson, Van Kannel Revolving Door Co., was then introduced by Mr. Eastman and spoke on Infiltration Through Doorways. Mr. Simpson's talk was illustrated by slides and was well received.

A rising vote of thanks was given both speakers for their very able presentations, according to the report of Secretary Erickson.

Mr. Smith, the retiring president, was presented with a golf bag as a token of esteem of the Chapter members, and he in turn, introduced the new officers and turned the gavel over to Pres. L. P. Hynes, who adjourned the meeting at 10:30 p. m.

March 11, 1937. The meeting of Philadelphia Chapter, at the Engineers Club, was called to order by Pres. W. F. Smith at 8:15 p. m. with an attendance of 40. The minutes of the previous meeting and the report of the Treasurer were read and approved.

A. E. Kriebel reported on the work of the Membership Committee, and plans for the April meeting were outlined by C. B. Eastman.

R. F. Hunger introduced the speaker of the evening, W. R. Heath, assistant chief engineer, Buffalo Forge Co., who spoke on Surface Cooling. Mr. Heath's talk, which was illustrated with slides, gave a comprehensive outline of the fundamentals of refrigeration in connection with air conditioning, and his lucid explanation of the psychrometric chart proved instructive and interesting. Mr. Heath was thanked on behalf of the members by R. C. Bolsinger.

According to the report of Secy. H. H. Erickson, President Smith adjourned the meeting at 8:45 p. m.

Dr. E. Vernon Hill Associated with Refrigeration and Air Conditioning Institute

Announcement has just been made by R. D. Smith, president, Refrigeration and Air Conditioning Institute, Chicago, Ill., of the appointment of Dr. E. Vernon Hill to the newly created post of director of research and education.

Dr. Hill, who assumed his new duties on April 1, will be responsible for outlining, coordinating and supervising preparation of the material used in the Institute training; will make contacts with engineering educational institutions; with research laboratories in the industry; and with engineering departments of air conditioning and refrigeration manufacturers.

Dr. Hill was president of the Chicago Commission on Ventilation in 1918, president of the Illinois Chapter of the A. S. H. V. E. in 1919, and president of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in 1920. Until recently he was editor of *The Aerologist*.

The Refrigeration and Air Conditioning Institute was founded in October 1934 for the purpose of training men to become skilled workers in the field of air conditioning and refrigeration. The program of training includes a shop preparatory training, given to the student by extension methods and completed in his own home, followed by a practical training in the Institute's shops and laboratories in Chicago.

Oil Burner Institute Elects Officers

During March 15-19, 1937 a meeting of the *Oil Burner Institute* and the National Oil Burner and Air Conditioning Exposition was held in Convention Hall, Philadelphia, Pa.

The following officers and executive committee were elected: *President*—C. N. Lockwood, Rock Island, Ill.; *Vice-Presidents*—J. J. Donovan, Bloomfield, N. J.; M. A. Powers, Detroit, Mich., F. H. Dewey, Detroit, Mich., Earl Marr, Columbus, Ohio, E. P. Bailey, Philadelphia, Pa.; *Executive Committee*—C. N. Lockwood, chairman, W. F. Brannan, ex-officio, J. J. Donovan, domestic burners, E. P. Bailey, industrial burners, M. A. Powers, domestic burners, J. H. O'Brien, industrial burners, F. H. Van Blarcom, distillate burners, and F. H. Dewey, domestic burners; *Directors*—J. H. O'Brien, Stamford, Conn., J. H. Van Sciver, Bethlehem, Pa., L. L. Knapp, York, Pa., S. C. Hope, Springfield, Mass., J. A. Lattner, Cedar Rapids, Ia., and A. E. Knapp, Detroit, Mich.

Death of E. W. Mandeville

In the death of Edgar W. Mandeville, Sr., which occurred on March 31, 1937, the Society lost a Member who had been associated with the A.S.H.V.E. for nearly a quarter of a century. Mr. Mandeville had been ill for some time and passed away at his home, 1171 East 37th St., Brooklyn, N. Y.

Born at Plains, Pa., on July 2, 1874, Mr. Mandeville was brought up in Elmira, N. Y., and he attended Sibley School, Cornell University. He was associated with the Herendeen Manufacturing Co., Geneva, N. Y., from 1899 to 1909 as engineer and salesman. Since 1910 he had operated his own business as the E. W. Mandeville Co., heating and ventilating contractors, located in Brooklyn, N. Y.

Mr. Mandeville was a member of Kings County Lodge, F. & A. M., and he was a trustee of the Lincoln Savings Bank. He was a vestryman of St. Paul's Church, Brooklyn, where services were held on April 2.

Surviving Mr. Mandeville are his widow, Mrs. Martha Byrne Mandeville, two sons, Dr. Frederick Mandeville, Richmond, Va., and E. W. Mandeville, Jr., Syracuse, N. Y., and a brother, Hubert C. Mandeville, to whom the Officers and Council of the Society have extended their sympathy in their bereavement.

Macy S. Good Dies

Macy S. Good, assistant sales manager, C. A. Dunham Co., Chicago, Ill., died on March 3, 1937, after a short illness, at the age of 61. Mr. Good was a Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for many years and he was well known and highly respected throughout the heating and ventilating industry.

Mr. Good was born at Richmond, Ind., and he attended Purdue University. His shop and drafting experience was obtained with the Brooks Locomotive Works, Dunkirk, N. Y., and with the Illinois Central System. From March 1903 to June 1905 he was with Fitz-Hugh Luther, dealers in locomotives and cars, in the capacity of salesman, shop superintendent, and buyer, and from June 1905 to August 1907 he was associated with White-Good Steel Co., brokers in high carbon and tool steels. In August 1907 he entered the paint and varnish manufacturing field in business for himself under the firm name of the Good Paint Co., later the Good Products Co. This was consolidated in November 1913 with the Nubian Paint and Varnish Co.

In January 1921 Mr. Good joined the C. A. Dunham Co. as sales manager for the Chicago district and his association with this company continued until the time of his death.

Mr. Good is survived by his widow and daughter, to whom the Officers and Council of the Society extend their sincere sympathy.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 35 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by May 15, 1937, these candidates will be ballotted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BAENDER, F. G., Pres., Thermo Air Cond. Inst., Inc., Los Angeles, Calif.	E. O. Eastwood H. M. Hendrickson	W. M. Sawdon C. A. Bulkeley
BEAULIEN, A. A., Utilization Engr., Boston Edison, Boston, Mass.	D. S. Boyden W. T. Jones	R. M. Nee E. A. Dusossoit (*Non-Member*)
BIRD, G. L. H., Chief Engr., Refrigeration (Birmingham) Ltd., Birmingham, England.	R. E. W. Butt J. E. Linebaugh (*Non-Member*)	W. W. Crosweller (*Non-Member*) B. Lloyd-Evans (*Non-Member*)
CADY, E. F., Sales Engr., American Blower Corp., Buffalo, N. Y.	H. D. Hexamer S. P. Eschenbach	V. P. John Joseph Davis
CARLSON, C. V., JR., Engr., Sidles Co., Airtemp Div., Lincoln, Nebr.	M. J. Stevenson F. E. Prawl	Arledge Stokes L. L. Daubert
CHARTERS, W. A., Htg. Sales, Canada Foundries & Forgings, Ltd., Brockville, Ont., Can.	W. D. Gordon H. R. Roth	Thomas McDonald J. H. Fox
COVER, E. B., Sales Engr., York Ice Mach. Corp., East St. Louis, Illinois.	T. W. McMahon L. J. DuBois	E. E. Carlson H. W. Meinholtz
DULL, E. J., Engr.-Contractor, Baltimore, Md.	J. deB. Shepard J. E. Seiter	E. L. Crosby H. B. Prewitt (*Non-Member*)
FELDERMANN, WILLIAM, Mgr. Air Cond. Div., American Gas Accumulator, Elizabeth, N. J.	H. W. Fiedler A. P. Livar	T. J. Charles J. A. Heller (*Non-Member*)
GREENBERG, IRVING, Design & Estimating, S. Greenberg, New York, N. Y.	W. R. Zuhlke M. C. Giannini	R. D. Lambert A. A. Adler
GUNNELL, G. T., Chief Engr., Sunbeam Htg. & Air Cond. Co., Atlanta, Ga.	J. E. Maynard H. K. McCain	R. F. Hahn C. A. Olsen (*Non-Member*)
HANLEIN, J. H., Asst. Secy. & Treas., Weller Corp., Washington, D. C.	C. S. Stock Glegge Thomas	G. H. Heine (*A. S. M. E.*) B. P. Hessler (*A. S. C. E.*)
HART, F. D., Grad. Student, Cornell Univ., Ithaca, N. Y.	W. M. Sawdon C. O. Mackey (*A. S. R. E.*)	W. N. Barnard (*A. S. M. E.*) F. O. Ellenwood (*A. S. M. E.*)
HILDER, F. L., Chief Engr., Electric Furnace-Man, Inc., New York, N. Y.	H. F. Cox L. E. Bannon	W. M. Heebner C. H. Hall (*Non-Member*)
HINES, G. M., Chief Engr., A. & M. College of Texas, College Station, Tex.	F. E. Giesecke W. H. Badgett	E. G. Smith W. E. Long
HUGHES, F. S., JR., Estimator, Hardin Sash & Door Co., Ft. Worth, Tex. (*Reinstatement*)	H. K. Kibbie (*Non-Member*) K. V. Kibbie (*Non-Member*)	J. V. Russell (*Non-Member*) C. C. McNutt (*Non-Member*)
KELLEY, R. D., Pres.-Treas., Sunbeam Htg. & Air Cond. Co., Atlanta, Ga.	J. E. Maynard H. K. McCain	K. E. Gouedy C. A. Olsen (*Non-Member*)
LEWIS, H. F., Vice Pres., Harvey A. Dwight Oil Heat & Supply Co., Inc., Albany, N. Y.	August Lux (*A. I. A.*) W. A. Clifton (*Non-Member*)	W. J. Provoost (*A. I. A.*) A. I. Schimpf (*Non-Member*)
MAHON, F. B., Air Cond. & Vent. Specialist, Duquesne Light Co., Pittsburgh, Pa.	R. J. J. Tennant J. F. Collins, Jr.	E. C. Smyers T. F. Rockwell
MANDELL, T. P., Sales, Carrier Corp., Boston, Mass.	James Holt J. W. Millard	R. T. Kern T. J. Earley
MARTENS, E. D., Mgr. Mech. Dept., Thompson Starrett Co., New York, N. Y.	W. H. Driscoll L. L. Munier	Arthur Ritter F. R. Still
MCCAFFERTY, J. E., Dist. Engr., Petroleum Heat & Power Co., Boston, Mass.	W. A. McPherson H. B. Wiegner	H. E. Whitten John Turner
MCKINNEY, C. A., Design Engr., United Gas System, Houston, Texas.	W. E. Stark I. E. Rowe	M. L. Diver A. B. Banowsky (*Non-Member*)
MOFFAT, O. G., Sales Engr., Livingston Stoker Co., Ltd., Hamilton, Ont., Can.	H. R. Roth R. P. Allsop	J. H. Fox Thomas McDonald
PELOUZE, H. L., 2nd, Br. Mgr., C. A. Dunham Co., Richmond, Va. (*Reinstatement and Advancement*)	C. A. Dunham O. J. Prentice	C. A. Thinn C. M. Brigham

CANDIDATES

ROY, LEO, Sales Engr., The S. W. P. Co., Montreal, Que., Can.

RUMMEL, A. J., Industrial Engr., San Antonio Public Service Co., San Antonio, Texas.

SCHOLL, H. O., Student, Univ. of Illinois, Urbana, Ill.

SHEPARD, J. DEB., Air Cond. Repr., Consolidated Gas, Elec. Light & Power Co. of Baltimore, Baltimore, Md. (*Advancement*)

SHEPHERD, C. B., Chemical Engr., E. I. du Pont de Nemours & Co., Wilmington, Dela.

SHERWOOD, L. T., Glass Technologist, Penna. Wire Glass Co., Dunbar, Pa.

SOETERS, MATTHEW, Consulting Engr., Smith, Hinchman & Grylls, Detroit, Mich.

TWIZELL, E. W., Partner, Connolly & Twizell Reg'r. (Htg. & Plbg. Contractors), Montreal, Que., Can.

WERNER, J. G., Supervisor, Htg. Sales, Equitable Gas Co., Pittsburgh, Pa.

WYLD, R. G., Executive Engr., Airtemp, Inc., Dayton, Ohio.

REFERENCES

	Proposers	*Seconders*
ROY, LEO	F. R. Barnsley F. G. Phipps	C. W. Johnson G. L. Wiggs
RUMMEL, A. J.	M. L. Diver R. W. Kotzebue	W. A. Ebert L. O. Vogelsang (*Non-Member*)
SCHOLL, H. O.	W. H. Severns A. P. Kratz	M. K. Fahnestock P. E. Mohn (*A.S.M.E.*)
SHEPARD, J. DEB.	J. E. Seiter E. L. Crosby	H. P. Gant A. P. Shanklin
SHEPHERD, C. B.	M. G. Kershaw C. R. Williams	R. E. Payne T. H. Chilton (*Non-Member*)
SHERWOOD, L. T.	R. A. Miller H. F. Smith (*Non-Member*)	E. G. Robinson (*Non-Member*) H. H. Blau (*Non-Member*)
SOETERS, MATTHEW	F. J. Feely R. K. Milward	J. S. Kilner W. G. Boales
TWIZELL, E. W.	J. W. O'Neill F. J. Friedman	G. E. Perras A. B. Darling
WERNER, J. G.	J. F. Collins, Jr. P. A. Edwards	E. S. Scanlon L. S. Maehling
WYLD, R. G.	I. C. Baker R. A. Gonzalez	F. A. Rodgers L. B. Miller

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ALLEN, C. V., Engrg. Mgr., Midwest Air Cond. Corp., St. Louis, Mo.

BALLMAN, W. H., Chief Engr., Krich Rodisco, Newark, N. J.

BENSON, B. C., Mgr. Sales Engrg., American Radiator Co., Chicago, Ill.

BLACKSHAW, J. L., Engr., Air & Refrigeration Corp., New York, N. Y. (*Advancement*)

BODMER, EMMANUEL, Oil Heat Engr., Ets. Dieny et Lucas, Paris, France.

BORAK, EUGENE, Chief Draftsman, Buensod-Stacey Air Cond., Inc., New York, N. Y.

BULKELEY, C. A., Chief Engr., Niagara Blower Co., New York, N. Y. (*Reinstatement*)

BURR, G. C., Sales Engr., Dail Steel Prod. Co., New York, N. Y.

CARBONE, J. H., Htg. & Vtg. Engr., L. J. Wing Mfg. Co., Baldwin, L. I., N. Y.

CRUMP, A. L., Sales Engr., Powers Reg. Co., Chicago, Ill.

DEAN, DAVID, Sales & Htg. Engr, N. Y. Plumbers Specialty Co., New York, N. Y.

EWENS, F. G., Demonstrator in Thermodynamics, University of Toronto, Toronto, Ont., Can.

FRANCIS, P. E., Asst. Sales Mgr., North Western Fuel Co., Minneapolis, Minn.

GREENLEAF, R. P., Air Cond. Engr., City Ice & Fuel Co., Cleveland, Ohio.

LINDSAY, G. W., JR., Design Engr., Frigidaire Div., Chicago, Ill.

MEYER, C. L., Mech. Sales, L. J. Wing Mfg. Co., New York, N. Y. (*Reinstatement*)

PAVEY, C. A., Dist. Mgr., B. F. Sturtevant Co., Detroit, Mich.

PICKETT, C. A., Sales Repr., Herman Nelson Corp., Moline, Ill. (*Reinstatement*)

RYERSON, H. E., Mgr. Air Cond. Div., Bryant Air Cond. Corp., Chicago, Ill.

SHAFFER, C. E., Research Engr., Koppers Co., Pittsburgh, Pa.

SLEMMONS, J. D., Br. Mgr., American Blower Corp., Worthington, Ohio.

SNAVELY, E. R., Chief Engr., Thermal Engrg. Co., Roselle, N. J.

VOORHEES, G. A., Vice Pres. & Engr., Century Htg. Service Co., Indianapolis, Ind. (*Reinstatement*)

WAGGONER, J. H., Research Engr., Owens-Illinois Glass Co., Muncie, Ind.

WHELAN, W. J., Purchaser & Estimator, Harrigan & Reid Co., Detroit, Mich. (*Reinstatement*)

YOUNG, H. J., Sales Engr., Young Radiator Co., Muskegon, Mich. (*Advancement*)

ASSOCIATES

BADARACCO, J. A., Owner, Badaracco Appliance Co., Mexico, Mo.

BIBER, H. A., Asst. Operating Engr., Mellon Natl. Bank, Pittsburgh, Pa.

BLACK, F. M., Chief Engr., Army Medical Center, Washington, D. C.

COOKE, T. C., Sales, Tomlinson Co., Inc., Durham, N. C.

CUTHBERTSON, M. W., Supt. of Bldg., Hardware Mutual Fire Ins. Co., Minneapolis, Minn.

DE VILBISS, P. T., Air Cond. Engr., Griffith Amusement Co., Oklahoma City, Okla.

EISELE, W. S., Owner, Ideal Heating, Buffalo, N. Y.

FOSTER, P. H., Mgr., Hudson Bay Plbg. Co., Flin Flon, Man., Can.

JEHLE, FERDINAND, Director Research Lab., Hoffman Specialty Co., Stamford, Conn.

JENNINGS, R. A., Chief Engr., Keith Mass. Corp., Boston, Mass.

LEE, J. A., Mgr. Refrigeration Div., Interstate Electric Co., Shreveport, La.

LEUPOLD, G. L., Sales Engr., Minneapolis-Honeywell Reg. Co., Cincinnati, Ohio.

LOUCKS, D. W., Supervisor, Com. Elec. & Steam Sales, Duquesne Light Co., Pittsburgh, Pa. (*Reinstatement*)

MULLOY, EDWARD, Mech. Supt., 131 Lincoln Rd., Inc., Brooklyn, N. Y.

SCHULER, W. B., Direct Factory Repr., Taco Heaters, Inc., Chicago, Ill.

SPENCER, E. D., Mgr. Commercial Dept., Brown Elec. Co., Oklahoma City, Okla.

SUTFIN, G. V., Sales Engr., American Blower Corp., Cincinnati, Ohio.

WAID, G. H., Sales Mgr., Scott Valve Mfg. Co., Detroit, Mich. (*Reinstatement*)

WARD, H. H., Air Cond. & Htg. Engr., Meltzer Motors, Miami, Fla.

JUNIORS

BLEDSOE, RAYMOND, Design Engr., The Trane Co., La Crosse, Wis.

BROWN, J. S., JR., Sales Engr., Delco-Frigidaire Cond. Div., Dayton, Ohio.

DAMM, D. A., Air Cond. Engr., Gerken Heating Corp., Toledo, Ohio.

FRIEDLINE, J. M., Sales Engr., Sidles Co., Airtemp Div., Des Moines, Iowa.

HECHLER, SAMUEL, Asst. Mgr., Westchester Square Plbg. Supply Co., New York, N. Y.

HELLMERS, C. C., JR., Sales Engr., Airtemp Div., Sidles Co., Lincoln, Nebr.

HESSELSCHWERDT, A. L., JR., Instructor Mech. Engrg., Wayne Univ., Detroit, Mich.

IRWIN, R. R., Air Cond. Engr., York Ice Mach. Corp., St. Louis, Mo.

KIMMEL, W. G., Sales Engr. York Ice Mach. Corp., Omaha, Nebr.

LANDEWIT, C. J., Combustion Engr., Combustion Maintenance Co., New York, N. Y.

MANNING, C. E., Air Cond. Sales Engr., von Hamm-Young Co., Ltd., Honolulu, Hawaii.

MOORE, B. J., JR., Archt., Griffith Amusement Co., Oklahoma City, Okla.

PLUMMER, R. S., Asst. Supt., Franklin Heating Station, Rochester, Minn.

SCHMIDT, H. I., Experimental Engr., Fedders Mfg. Co., Inc., Buffalo, N. Y.

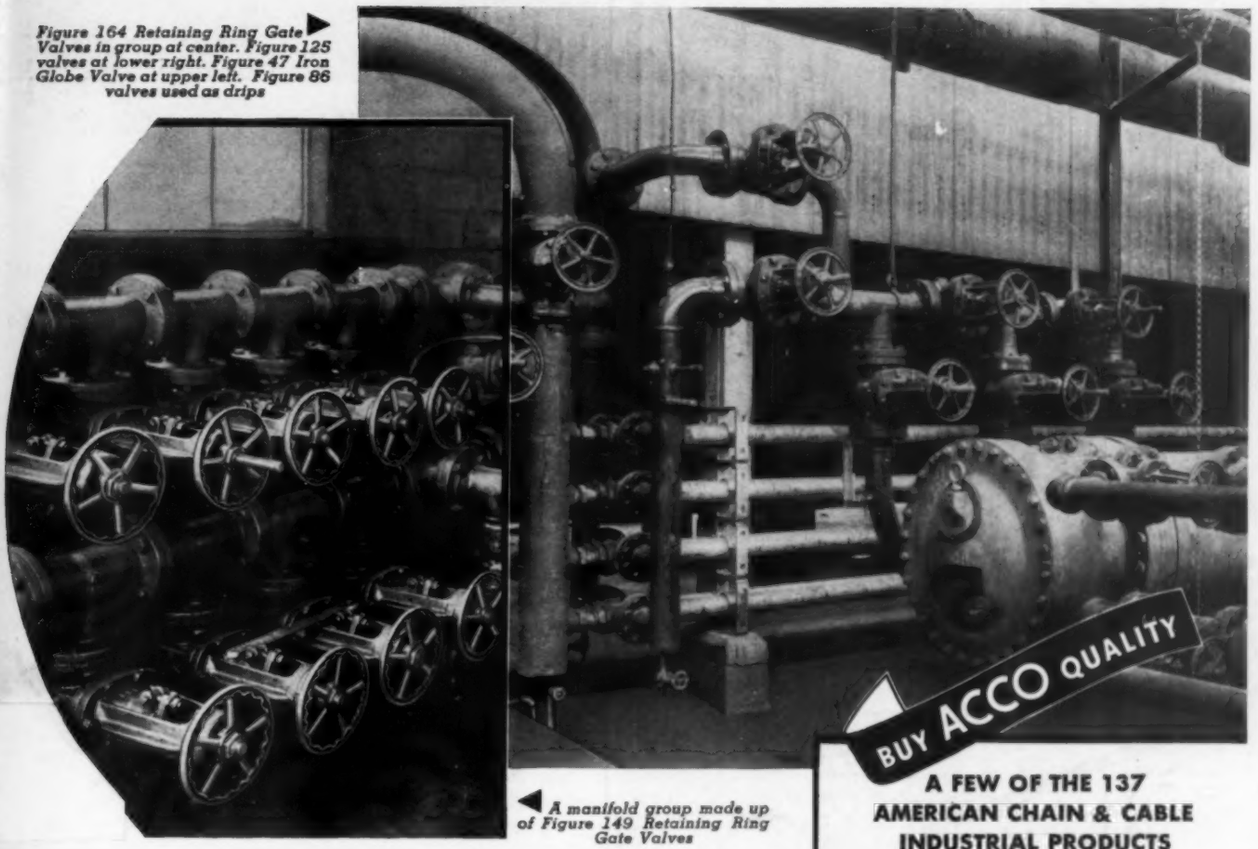
STUDENTS

HEDLUND, R. A., Student, Purdue University, West Lafayette, Ind.

STEINKE, B. J., Drafting, B. H. Steinke, New York, N. Y.

WOLIN, M. W., JR., Engr. & Draftsman, Typhoon Air Cond. Co., New York, N. Y.

Figure 164 Retaining Ring Gate Valves in group at center. Figure 125 valves at lower right. Figure 47 Iron Globe Valve at upper left. Figure 86 valves used as drips



A manifold group made up of Figure 149 Retaining Ring Gate Valves

BUY ACCO QUALITY

Specified because of *proved*

DEPENDABILITY

● When iron body gate valves are required, men with long valve experience think at once of Reading-Pratt & Cady. Many of them know that Pratt & Cady patented and introduced the retaining ring type of valve construction over fifty years ago. All know that when they specify Reading-Pratt & Cady Retaining Ring Gate Valves they will get dependable service.

Constant development and improvement over the past fifty years assure that Reading-Pratt & Cady Retaining Ring Gate Valves will give long and economical service under exacting modern requirements.

Literature containing full details will be sent upon request.

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In Business for Your Safety

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(DOMINION CHAIN COMPANY, Ltd., in Canada)
Weed Tire Chains • Welded and Weldless Chain • Malleable Castings • Railroad Specialties

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FORD CHAIN BLOCK DIVISION
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HAZARD WIRE ROPE DIVISION
Green Strand Wire Rope • "Korödless" Wire Rope • Preformed Spring-Lay Wire Rope • Guard Rail Cable

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Wrought Iron Bars and Shapes

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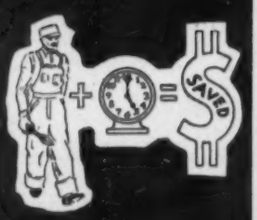
READING·PRATT & CADY VALVES

PERFECT INSULATION

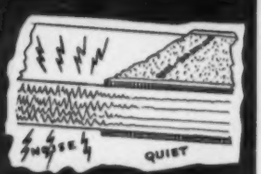
DUX-SULATION

Combining Efficient Insulation, Appearance and Long Wearing qualities with Flexibility, Easy Installation and Low Cost.

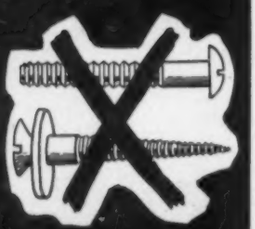
ABSOLUTE FLEXIBILITY



LABOR SAVING



SOUND ABSORPTION



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Inland Container Company Job. Indianapolis. Ind. Westerlin & Campbell Co., Distributors for York Ice Machinery Corp., Air Conditioning Contractor. J. R. Kreglo, Insulation Contractor.

DUX-SULATION, made of fine felted fibre with a re-inforced outer surface that has an attractive texture finish, will insulate perfectly. Applied without the use of bolts, lugs or screws it eliminates the possibility of opening duct seams by pounding. It assures a perfect job—everytime. When you join pieces of Dux-Sulation the texture of the felted fibre is such that they become practically one. To cover these joints we furnish Dux-Sul-Tape a waterproof friction type tape.

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Book Reviews

Hydraulic Institute Publishes Pump Data

"Standards," published by the Hydraulic Institute, 90 West St., New York, N. Y., comprising seven separate sections in a paper cover, 8½x11 in. 7th ed. Price (complete set), $1.00. Prices of the separate sections range from 25c to 50c..

Pursuant to its aim of being of service to engineers, buyers, and users of pumps, the *Hydraulic Institute*—comprising many leading manufacturers of displacement and centrifugal pumps—has compiled the pertinent engineering and technical data and recommended standard definitions, terms and practices published in this book. The Institute believes that general adoption of the recommendations will be of mutual advantage to users, the engineering profession, and pump manufacturers.

This book is made up of seven separate sections on general information, centrifugal pumps, rotary pumps, reciprocating pumps, deep well turbine pumps, test code for centrifugal and rotary pumps and engineering data. The deep well pump section has not yet been printed but will be available shortly. As an example of the extent of the information given, the following subjects are treated in the centrifugal pump section: Definitions of types and parts; standards of volume, head, power, and efficiency guarantees; specific speed and suction lift; maximum permissible capacity curves for various suction conditions; instructions for installing and operating centrifugal pumps; and special applications.

The 36 p. data section includes friction tables for w
on resistance of valves and fittings, viscosity, friction of oil in pipe lines, friction of paper stock in cast iron pipe, flange dimensions, steel flange fittings, materials for pumping various liquids, properties of water at various temperatures, etc.

Practical Boiler Firing

"Practical Boiler Firing," by H. C. Armstrong and C. V. Lewis, Fuel Officer and Power Station Superintendent, respectively, to Messrs. Thos. Firth and John Brown, Ltd. Published by J. B. Lippincott Co., Washington Sq., Philadelphia, Pa. 1st ed. 1936, 132+ xii pp., 5½x7¾ in., clothbound. Price, $1.50.

"He who fires must think" say the authors, and it is their purpose in this little book to teach scientific methods, but in a way which the practical worker can readily understand without being discouraged by technical terms "beloved by the academician."

Nine chapters (on combustion, fuel, stoking, boilers, boiler mountings, measuring instruments, feedwater treatment, boiler efficiency, and steam mains) are included.

Boiler Reference Book

"The Pape-Swift Boiler Reference Book, 1937," published by John S. Swift Co., Inc., 105 S. 9th St., St. Louis, Mo. Planographed, 333 pp., 5¼x8¼ in., paper cover. Price, $1.00 plus 10c postage.

This is the fourth of the Pape-Swift boiler reference books, which cover in all over 900 different models of boilers of more than 80 different manufacturers. A master index to the four books is furnished with it.

Offered as a quick and ready reference means for determining the ratings of boilers, the tables given are reproduced directly from data furnished by the boiler makers. The 1937 edition covers boilers of some 30 manufacturers, and includes specifications and line drawings of newly designed boilers recently put on the market.

Sections are twins. Grates are twins. Base is in sections. Has a three times back and forth fire travel. The long fire travel that makes its short fuel bill.

Lincoln Looked At It Exactly The Same

YOU'LL recall his being asked about a certain book, whether it was a needful one. He replied: "Any one who needs that book, that's the book they need."

The same in a sense, is true of this Twin Sectional Boiler of ours. If you have a replacement job, it can be installed without any tearing out, as no part of the boiler is so large it can't easily be passed through any average door.

If you prefer a cast iron boiler to a steel one, as so many do, then this boiler deserves your consideration.

If economy is a deciding factor, then this boiler meets that requirement. Its *long* fire travel makes a short fuel bill.

The fact that it is made by a concern that for over three quarters of a century, has been making boilers, may or may not interest you. It's at least worth mentioning. It's some assurance that in the years to come, repair parts will be available.

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Patented

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BETTER SERVICE with Randall BEARINGS

The Randall D.R.O.R. (double reservoir oil return) precision bored pillow blocks are rapidly becoming standard equipment in the air conditioning field. With a minimum of attention, they will operate practically a life time. They are self-aligning, self-lubricating, require little attention and operate with minimum running torque.

Cut-away section of ball of D. R. O. R. and Flange Type Pillow Block showing Double Reservoir Oil Return feature.

Since the introduction of the D.R.O.R. sales have nearly doubled monthly.

Blowers equipped with Randall Bearings deliver air throughout the system WITHOUT METALLIC BEARING NOISE. Write for our new Pillow Block catalog or outline any special bearing problems you have.

RANDALL FLANGE-MOUNT PILLOW BLOCK

Randall Flange D.R.O.R. Pillow Blocks can be easily mounted or incorporated in the design of cabinet type air conditioning equipment.

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Dept. 510 609 W. Lake St., Chicago, Ill.

Air Conditioning—Insulation

"Air Conditioning—Insulation," by J. Ralph Dalzell, Head, Architectural Engrg. Dept., American School, and James McKinney, Educational Director, American School. Published by American Technical Society, Drexel Ave. at 58th St., Chicago, Ill. 1st ed. 1937. 301 pp., 5½x8½ in., clothbound. Price, $2.50.

Purpose of this book is to teach the principles and applications of insulation of buildings to retard heat losses and gains, prevent fire loss, control sound, prevent vibration, protect against termites, and protect against condensation. The authors look upon insulation as an individual science which is a means of making heating and air conditioning economically sound.

Subjects treated include physics of insulation, kinds of insulation, where insulation is used, transmission coefficients and tables, heating and cooling loads, and design of insulation. Data presented in the *A. S. H. V. E. Guide*, and other sources, have been freely drawn upon. The examples pertain principally to residential construction.

Estimation of Poisonous Gases in Air

"Estimation of Poisonous Gases and Vapors in the Air," by A. S. Zhitkova, Supervisor, Moscow Workers Chemical Laboratory, edited by S. I. Kaplun, Supervisor, Central Executive Institute of Work Sanitation, and translated under the direction of Joseph B. Ficklen, Chemical Engineer, Travelers Insurance Co. Published by Service to Industry, Box 133, West Hartford, Conn. 1936. 198 + xiii pp., 6 x 9 in., clothbound. Price, $3.00.

This is a practical manual on a relatively new field of applied chemistry, the detection and estimation of substances in air for industrial health purposes. Besides the methods of detection and estimation given, fundamental data on each material and concentrations deemed to be objectionable are presented. For material, it has been possible to draw upon the experience of the largest industrial hygiene laboratory in the U. S. S. R.

In addition to chapters on selection of the air sample and determination of substances in the air of industrial establishments, a chapter is devoted to each of the following—carbon dioxide, carbon monoxide, sulphur dioxide, hydrogen sulphide, carbon bisulphide, nitric acid and the oxides of nitrogen, ammonia, chlorine, hydrogen chloride, hydrogen fluoride, hydrogen cyanide, arsenic and arsine, metals (general), zinc, mercury, lead, organic materials (general), methane, acetylene, formaldehyde, acrolein, acetone, alcohols, phenol, esters, ethyl ether, benzine, aniline, nitrobenzene, and turpentine.

An appendix is devoted to technical gas analysis, and tables of toxicity of gases and vapors, and specific gravity of gases and vapors, are presented.

Forced Air Heating

"Forced Air Heating," by Platte Overton. Published by Keeney Publishing Co., 6 N. Michigan Ave., Chicago, Ill. 2nd edition, 1937. 215 pp., 6 x 9 in., clothbound. Price, $2.00.

Written to serve as a handbook of design procedure, the new edition of this well-known book has been revised to present the material in the order in which the problems of design arise in the preparation of plans for residential forced air heating and air conditioning systems. It is the author's purpose to explain the middle course between too much time spent in the preparation of plans, and too little time—the former arising because the designer is not sure his procedure accounts for all factors, and the latter characterized by the use of a few rules-of-thumb applied haphazardly.

Chapters are included on a suggested data sheet, design procedure, sizing apparatus, infiltration, registers and returns, multiple registers, return air systems, register temperatures, temperature drop in ducts, mechanical code for forced air heating, sizing ducts by friction chart method, sizing extra long pressure losses, velocities, proportioning losses in heating systems, branch take-offs, volume dampers, bonnets and plenums, casing baffling, chimneys and drafts, control, and contract and specifications.

REX-WELD "Super-Service"
VIBRATION ABSORBERS
for Air-Conditioning Machinery
● Here's the ideal solution to the compressor noise problem! Installed in the suction and discharge lines, these Rex-Weld Flexible Metal Tubing Units effectively dampen the vibration and put an end to the annoying hammering—surely, safely and economically . . . Produced by an exclusive patented process which forms the flexible tubing corrugations without weakening the special, non-porous, non-crystallizing bronze alloy. Supplied complete with capped ends ready for sweating onto standard water tubing. Write for Bulletin RW-4 and Data Sheet. Send us your specifications for prices.
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(Formerly Chicago Tubing and Braiding Company—Est. 1902)
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Cross section photograph showing how New Trimo Sawing Vise Set will hold shortest pieces of tubing and give true cut even when shortening measure is only fraction of an inch. Easy to line up cutting slot in Vise with cutting mark on tubing.

SAWING VISE SET

PAYS FOR ITSELF ON FIRST JOB

This standard Trimo product actually pays for itself on its first job of cutting thin wall copper tubing by

1. Making every cut 100% square
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Order a set to-day and take all the grief out of thin wall copper tubing jobs.

TRIMO

In the wrench industry the standard is Trimo. Always identified by the Red Tag attached to each Trimo Pipe Wrench.

TRICROME CUTTER WHEELS

Thin Blade Special Analysis Steel—silvered to resist rust. Identified by Red Hub. Fit Trimo, Barnes, Saunders Cutters.

TRIMONT MFG. CO., INC.
Roxbury (Boston), Mass.

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Air conditions 20 theatres in Baltimore alone—and each installation is paying for itself in 2 to 3 years.

Earning such returns is not unusual with Frick Refrigeration: restaurants increase profits up to 300 per cent; dairymen get Grade A prices for their milk; ice cream and quick freezing plants save up to 34 per cent in horsepower consumed; one-man ice plants produce and store up to 22 tons of ice per day; 50 other businesses find it equally indispensable.

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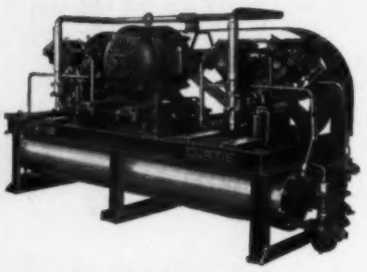
FRICK CO.
DEPENDABLE REFRIGERATION SINCE 1882

Performance Proves the Quality of

CURTIS CONDENSING UNITS



● Day after day, in thousands of air conditioning and refrigeration installations, the superior quality of Curtis design and construction is being proven beyond a doubt. Dependable, efficient, care-free performance is a characteristic of every one of the 86 Curtis Units, from 1/6 HP to 30 HP, air and water cooled.

Curtis quality is the result of 83 years of experience, of advanced engineering and precision manufacturing with 43 years specializing in building fine compressors.

Curtis engineering created the patented "Centro-Ring" method of positive pressure lubrication with no wearing parts. Curtis designing includes Timken Tapered Roller Main Bearings, water jacketed compressor heads and cylinders, drop forged heat-treated crankshafts and rods, balanced sylphon bellows seal, built-in oil separator with automatic return and automatic water valves, and precision manufacture throughout.

There's a Curtis condensing unit for every air conditioning and refrigeration need. Curtis quality guarantees superior performance.

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In Canada:
Canadian Curtis Refrigeration Company, Ltd.
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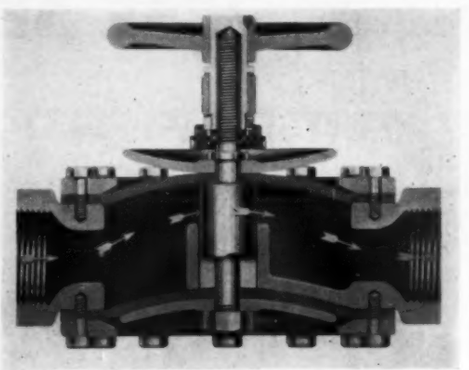
CURTIS

Equipment Developments

For your convenience in obtaining more information about any of this equipment, see coupon on page 160. Add the new products and companies listed here to your Directory Section which you received in your January, 1937, HEATING, PIPING AND AIR CONDITIONING and thus keep your records of sources of supply up to date throughout the year . . . Single asterisk (*) indicates equipment not listed in Directory Section; double asterisk (**) equipment and manufacturer not listed.

Balanced Diaphragm Valve Closes Tight

No. 1215—New balanced diaphragm valve has the feature of closing off tight repeatedly, in service on fluids or gases containing foreign matter, the manufacturer states. Being balanced, it will open or close easily. There are no bellows or stuffing boxes, thus preventing friction. When used on throttling service, there are no bad effects from wire drawing, it is said.

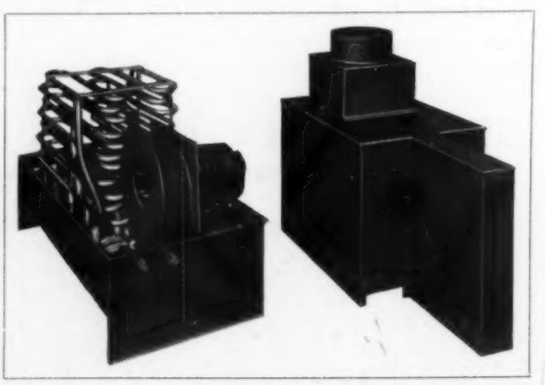


This valve need not be removed from the line to change the diaphragm, which can be supplied of various compositions to take care of air, water, gas, acids, or alkalis. Made of either ferrous or non-ferrous materials, it can be suitably lined for acids and alkalis, hot and cold. It will operate in any position, will not damage from freezing, and can be either hand or motor operated, says the maker. Made in sizes of 2 in. and larger.—McAlear Mfg. Co., 1901-1909 S. Western Ave., Chicago, Ill.

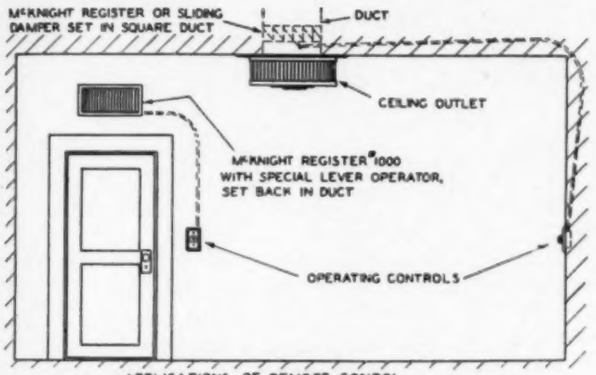
Evaporative Condenser Incorporates New Feature

No. 1216**—New "Water Saver" (evaporative condenser) eliminates the use of pumps, spray nozzles, etc. While the principle of operation is the same as other evaporative condensers, the means used to convert the solid water into a fog-like spray constitutes a radical change.

In the base of the unit is a water reservoir in which the height of water is automatically controlled by a float valve. A circular flange attached to the blower fan extends into the water and, rotating at high speed, picks up water and converts



ONE OF THE TOUGHEST PROBLEMS of AIR CONDITIONING SOLVED!

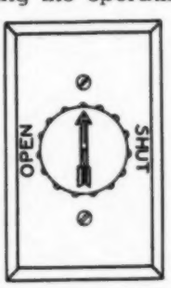

McKNIGHT REGISTER OR SLIDING DAMPER SET IN SQUARE DUCT
DUCT
CEILING OUTLET
McKNIGHT REGISTER #1000 WITH SPECIAL LEVER OPERATOR, SET BACK IN DUCT
OPERATING CONTROLS

APPLICATIONS OF REMOTE CONTOL

SEVERAL SUGGESTIONS SHOWING THE APPLICATION OF THE REMOTE CONTROL 1ST - REMOTE CONTROL CONNECTED WITH LEVER OPERATED McKNIGHT REGISTER BOX PLACED OVER A ROOM DOOR. 2ND - REMOTE CONTROL CONNECTED WITH A SPECIAL CEILING OUTLET DAMPER

REMOTE CONTROL

Tuttle & Bailey Remote Control allows the occupant of a given room in a hotel, large apartment house, office building, etc. to control the volume of air supply at a convenient point within the room which may be considerably removed from the volume damper! It consists of a control box containing the operating mechanism which is installed in the wall at a convenient height. A Knob mounted on an attractive escutcheon plate similar to an electric switch, serves as the operator. A flexible cable connects the operator mechanism and the valve damper. The Tuttle & Bailey McKnight Register box serves ideally as a volume control damper, and when combined with the Flexair Grille which allows for air deflection adjustable at the outlet, the installation becomes the most easily adjustable and flexible system in Air conditioning work.



"THE STANDARD OF COMPARISON"

T & B

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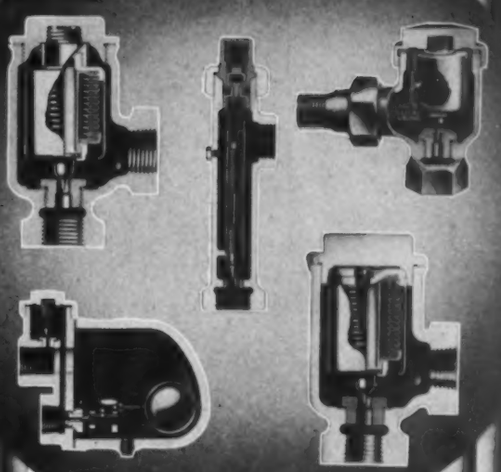

We'll send you a Sarco Steam Trap on approval at our expense. Try it for 30 days at the most troublesome point on your steam heated equipment. Then decide whether you want to keep it. If you prefer to return the trap, do so at our expense and that settles everything.

Thousands of Sarco Steam Traps have been sold on this 'show-me' basis. Surely it is worth your while to find out, under the protection of this buy-it-only-if-you-like-it basis, whether Sarco Steam Traps will save money for you as they have for many thousands of others.

You can't lose on this offer. So write today for Catalog 0-48 and a Sarco Steam Trap, stating size and pressure, or simply mail the coupon.

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SARCO STEAM TRAPS

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Without any obligation, you may
☐ Send one.....inch Sarco Steam Trap for.....lbs. pressure on 30 days' free trial.
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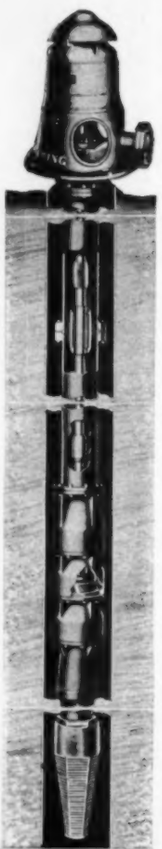
(A salesman of expensive air cooling equipment meets a typical satisfied owner of an economical well water cooling system).

Deming Deep Well Turbine Pumps have many features that recommend them for summer air cooling as well as for general water supply requirements. • They are water lubricated. (No oil or grease to contaminate the water). Adjustable impeller clearance makes it easy to sustain high efficiencies over a long period of service. Other features include cutless rubber bearings and polished stainless steel bearing sleeves which add to the high standard of performance and long life of these pumps.



• DEMING DEEP WELL TURBINE PUMPS *are illustrated and explained in this Free catalog. Send for your copy now.*

DEMING
Pumps
THE DEMING CO. • SALEM, OHIO

it into a fine mist which is distributed over the refrigerant coils. This flange and the blower constitute the only moving parts in the unit. The coil surface is arranged in banks in such manner as to allow for easy cleaning.

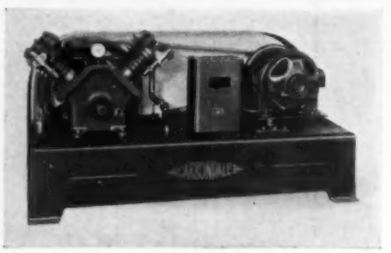
Discharge of solid water is prevented by means of an eliminator assembly on the air discharge outlets. Provision is made to supply water for water cooled compressor heads by means of an accumulator discharging water through the head and returning to the "Water Saver" to be re-used.

Unit is available for compressor capacities of from 1½ to 25 tons.—Peerless of America, Inc., 515 West 35th St., Chicago, Ill.

Condensing Units Designed for Air Conditioning Service

No. 1217—New self-contained low pressure refrigerating units are designed for either "Freon" or methyl chloride and are recommended by the maker for both air conditioning service and general refrigerating purposes.

The compressor unit is compact, mounted on a welded steel base, and requires small floor space and low head room. Suction and discharge valves are the light weight, quick acting, "feather" type. V-type compressors, from four to eight cylinders depending upon the capacity, are made of a special metal. Bearings and eccentric rod are bronze.



Electric motor drive to the compressor is the V-belt type. A self-adjusting motor bedplate maintains proper belt tension at all times, except on the eight cylinder unit which has adjusting screws for the motor baseplate.

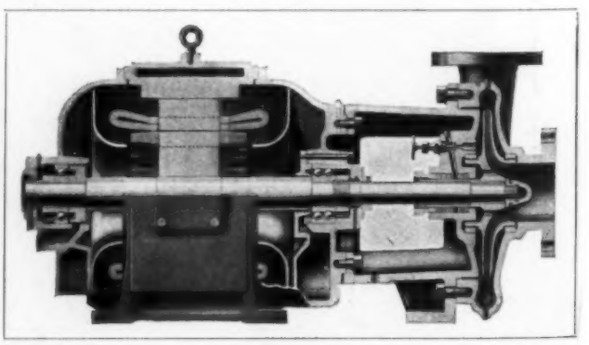
Safe operation is assured by interlocking the motor starter and the high pressure cut-out. Starting is accomplished by means of pressure or temperature changes, and uses the usual automatic starting features.

The receiver is integral with a shell type condenser which is fitted with copper finned condensing tubing and is of welded construction throughout.—Carbondale Div., Worthington Pump and Machinery Corp., Harrison, N. J.

New Centrifugal Pump Units Announced

No. 1218—A new line of motor driven centrifugal pumps for capacities of from 5 to 1200 gpm and heads of 10 to 230 ft has been announced. The pump is mounted upon the end bell of an electric motor, the two forming a compact and self-contained unit, with but one shaft and two bearings.

The motor-mounted pump is shipped ready for immediate connection to piping and power supply line and can be installed with the shaft horizontal, vertical, or inclined, or the unit can be mounted upon a hand truck or suspended from a sling. It is suitable for handling cold or hot water, brine, chemical solu-



1 set of dies

threads 4 sizes of pipe

Instead of 16 separate dies usually needed to thread 1", 1¼", 1½" and 2" pipe, this RIDGID No. 65R threader threads them all perfectly with one set of 4 dies, that stay in the threaders, can't get lost.

Move the change plate to the size of pipe and the threader is ready for work. Even the new style workholder sets to size with a twist of a gauge ring—put it on the pipe and tighten with one screw.

You'll like these and other features of this remarkable tool. Ask your Jobber to show you a RIDGID No. 65R.

THE RIDGE TOOL CO.
ELYRIA, OHIO, U. S. A.

RIDGID PIPE TOOLS

75% of your wrench repairs are stopped when you use RIDGID with the unconditionally guaranteed housing.

DELIVER COOL AIR

MORE EFFICIENTLY • MORE ECONOMICALLY

AIR CONDITIONING systems operate more efficiently—at lower cost—when cool air ducts are insulated with Novoid Corkboard. Properly applied to round, square, or rectangular ducts, this long-life material effectively resists the passage of heat and prevents condensation. It helps assure accurate control of temperature and lowers operating costs.

For insulating air-washing rooms, Novoid Corkboard is equally efficient, and prevents costly infiltration of warm air. Novoid is light in weight, structurally strong, easy to erect. For more detailed information, mail the coupon or write Cork Import Corporation, 330 West 42nd Street, New York City.

BRANCH OFFICES: Chicago, 400 West Madison Street; Boston, 248 Boylston Street; Philadelphia, 1524 Chestnut Street.

NOVOID CORKBOARD INSULATION

FOR YOUR CONVENIENCE

Cork Import Corporation
330 West 42nd Street, New York City

Please send me complete details of Novoid Corkboard and its application to air ducts.

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tions, extracts, juices, liquors, oil and gasoline, etc., in buildings and in various industrial and process plants.

The pump casing is of cast iron, while the impeller, wearing ring, shaft sleeve, stuffing box gland and lantern ring are of bronze. Special materials can be supplied where required. All pump parts can be removed without disturbing the motor.—De Laval Steam Turbine Co., 300 Nottingham Way, Trenton, N. J.

Controls Bacteria in Air Conditioning

No. 1219—A new material known as "Oakite Airefiner" has been developed to solve the problem of bacteria and slime control in air washing systems. When added to the recirculating water used to wash or scrub air, it keeps the water sterile and prevents the growth of slime and algae deposits in the system, according to the maker.

It is said to be completely soluble, to transmit no odor to water or air, and to provide a stable, colorless solution that is safe and non-toxic. It is non-corrosive to metal surfaces, helps prevent water scale formation.—Oakite Products, Inc., 22 Thames St., New York, N. Y.

Electrically-Operated Flow Meter Offered

No. 1220—A new electrically operated flow meter, known as the "Synchro-Meter," consists basically of the transmitting unit illustrated which is operated by a Ledoux bell flow measuring mechanism of the type used in standard mechanically operated fluid meters, and a receiver consisting of the indicating, recording and integrating instruments. The transmitter is located at the point of measurement of the variable and the receiver is located at a place or places convenient for obtaining readings from the instruments as frequently as desired.

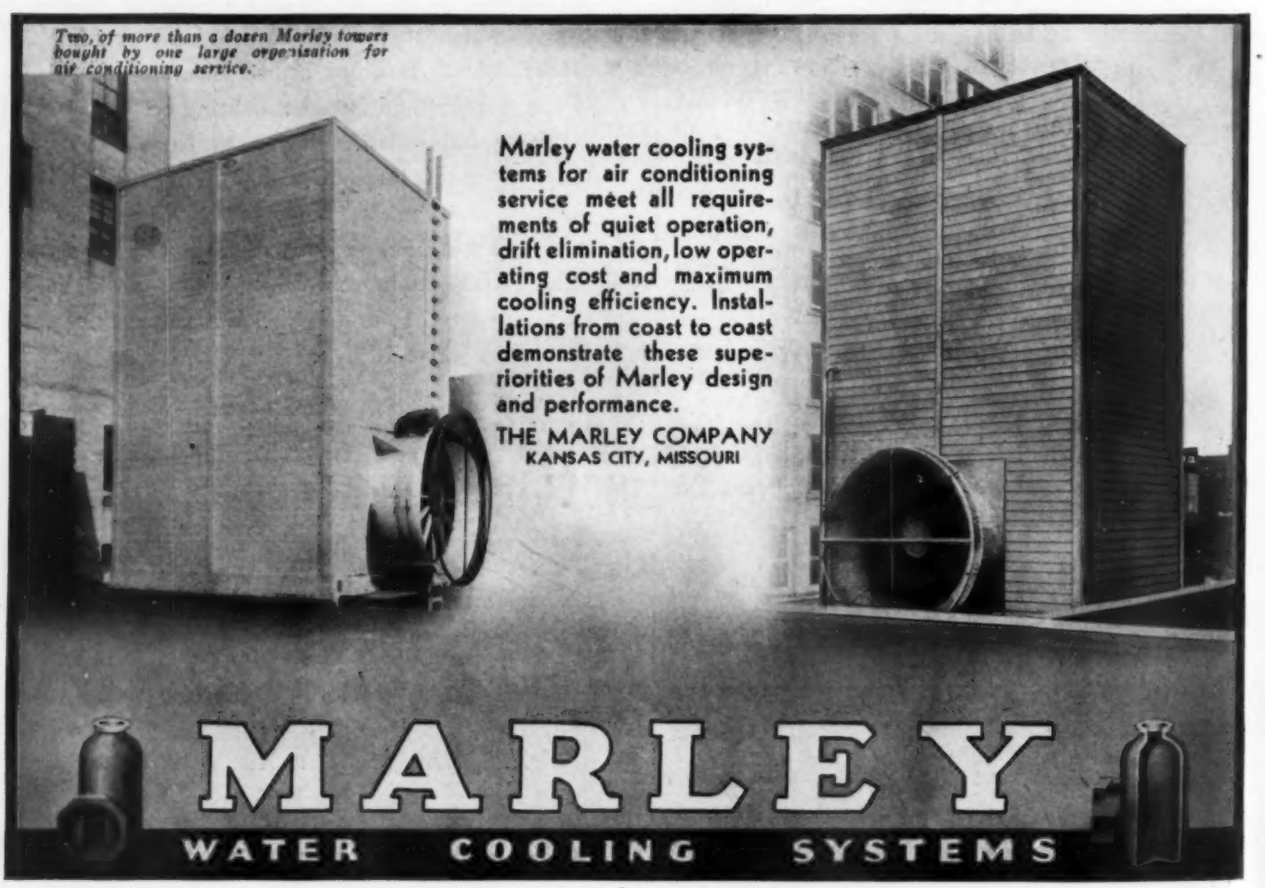
A number of advantages characterize the meter, according to the manufacturer. For example, the transmitter places only a negligible retarding force on the measuring device. Furthermore, the torque developed to operate the receiving element is very large and does not send any reaction back to the transmitter. No adjustments are necessary at the transmitting end; it may therefore be located in a relatively inaccessible place. Several receivers may be operated from one transmitter.

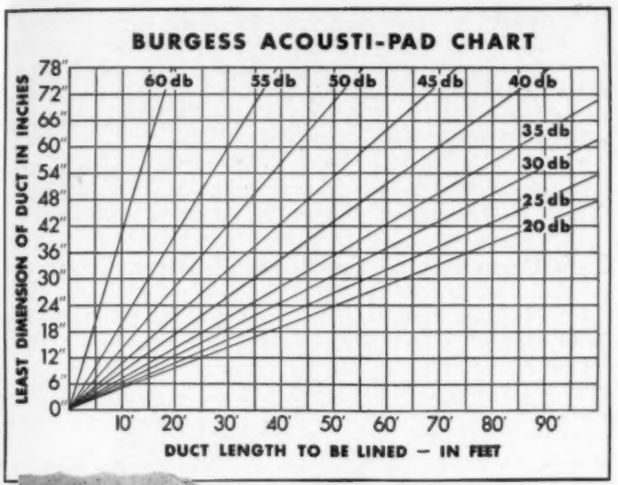
Three wires are required to connect the transmitter to the receiver. With the use of No. 12 wire the transmitter may be located approximately one mile from the receiver. If larger wire is used, a greater distance is allowable providing the loop resistance does not exceed 20 ohms. The use of alternating current at from 95 to 125 volts, and at a frequency of 25, 50 or 60 cycles, is required.—Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio.

Valve Controls Steam for Heating, Process

No. 1221—New valve of diaphragm type is designed to control the flow of steam through a full range of pressures from 10 lb to as low as 28 in. of vacuum.

A spring attached to one side of the diaphragm on the low pressure side of the valve can be placed under tension or com-


Two, of more than a dozen Marley towers bought by one large organization for air conditioning service.
Marley water cooling systems for air conditioning service meet all requirements of quiet operation, drift elimination, low operating cost and maximum cooling efficiency. Installations from coast to coast demonstrate these superiorities of Marley design and performance.
THE MARLEY COMPANY
KANSAS CITY, MISSOURI
MARLEY
WATER COOLING SYSTEMS


BURGESS ACOUSTI-PAD CHART
LEAST DIMENSION OF DUCT IN INCHES
78" 72" 66" 60" 54" 48" 42" 36" 30" 24" 18" 12" 6" 0"
60 db
55 db
50 db
45 db
40 db
35 db
30 db
25 db
20 db
10' 20' 30' 40' 50' 60' 70' 80' 90'
DUCT LENGTH TO BE LINED — IN FEET

It's easy to figure

the footage of Burgess Acousti-Pad for absorbing duct noises

Burgess Acousti-Pad, the perforated, metal-faced acoustic duct lining that blots up air-borne noises as efficiently as a blotter absorbs ink, is the perfect lining for modern air conditioning systems. There is no guesswork in estimating Burgess Acousti-Pad requirements; accurate data sheets are now available to architects and engineers. Use them and the final results will meet specifications.

Burgess Acousti-Pad is more than efficient—it is safe. Its perforated metal facing acts like the screen of a miner's lamp. It protects the sound absorbent material from flame—an important consideration in ventilating systems.

Also send for Bulletin 127 on the Burgess Model 202 Acousti-Booth for hotels, stores, and other public places. It is the first doorless booth designed for telephone service.

Burgess Battery Co., Acoustic Div.
111 W. Monroe St., Dept. H
Chicago, Ill.

Licensed under C. F. Burgess Laboratories, Inc., Patents

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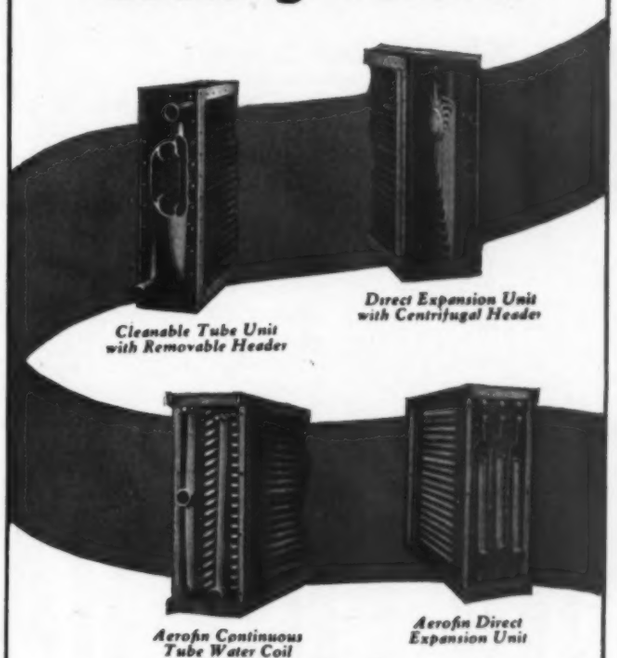
For Dependable
COMFORT COOLING
Specify
AEROFIN
Cooling Surface



Cleanable Tube Unit with Removable Header

Direct Expansion Unit with Centrifugal Header

Aerofin Continuous Tube Water Coil

Aerofin Direct Expansion Unit

A COMPLETE LINE
For Every Requirement

AEROFIN CLEANABLE TUBE UNITS: For cooling only. Headers removable to permit cleaning of tube interiors. Recommended where sediment or scale-forming chemicals are present in cooling water.

AEROFIN DIRECT EXPANSION UNITS: Centrifugal header type. Recommended where one valve control of unit is desired.

AEROFIN CONTINUOUS TUBE WATER COILS: For air cooling by circulation of cold water through tubes. Made for horizontal or vertical air flow.

AEROFIN DIRECT EXPANSION UNITS: Row control type. For use where cutting rows of tubes on or off in direction of air flow is desired.

Ask Newark for complete descriptive literature and data or call on the nearest district office for technical assistance.

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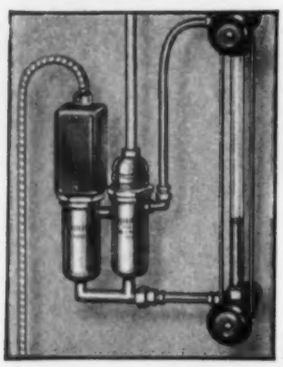
pression; an indicator shows the spring condition. Turning the handwheel at the bottom of the assembly adjusts the valve to control pressures above, at, or below atmosphere. There are no stuffing boxes or other sealing devices used, so that the valve instantly responds to the slightest variation in load, according to the maker. The diaphragm maintains set pressure or vacuum on the delivery side of the valve under fluctuating inlet pressure.

The diaphragm is of composition or metal for given conditions. Valve seat is double. Valve is made with iron body, bronze trim, in sizes to 10 in.—Illinois Engineering Co., 2035 S. Racine Ave., Chicago, Ill.

Heating Boiler Protector Improved



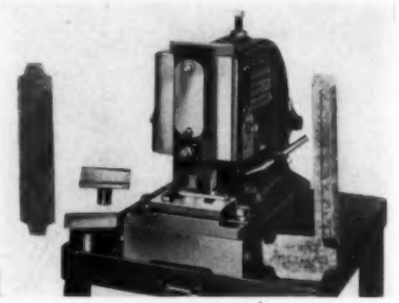
No. 1222—Insurance against low water or excessive pressure in automatically fired low pressure steam heating boilers is provided by improved "Micro Water Boy" boiler protector, a combination unit consisting of a safety feeder, boiler protector and the necessary fittings for connection to the water glass.

The "Micro" switch is used as a circuit breaker for both low water and excessive pressure. Normally in an open position, it is only closed when the float raises and snaps it by compound leverage. It automatically throws out when the float recedes. A bronze diaphgram in the float chamber prevents water or steam leaking into the switch box; when subjected to excessive steam pressure, it moves upward, breaking the contact with the switch until the pressure recedes.

The feeder is installed away from the boiler to prevent corrosion.—Maid-O'-Mist, Inc., 180 N. Wacker Dr., Chicago, Ill.

Kick Press Device Cuts Cost of Drive Cleats

No. 1223—Development of a special notching punch and die to be used on the manufacturer's standard No. 28 kick press for reducing the cost of producing drive cleats (also called "drive slips"), which are used in fastening duct sections together, was announced last month. By using this tool, a material saving in the production of these cleats can be realized, states the maker.



As shown, the punch cuts two special tapered notches at the end of the galvanized cleat stock in one operation, after which the sides are folded in (by other means) to form the finished cleat. Also shown is a shearing punch and die for cutting off straight stock to the desired lengths, this punch and shoe being easily substituted for the notching unit.—Whitney Metal Tool Co., 110 Forbes St., Rockford, Ill.

NO FIRE HAZARD HERE!

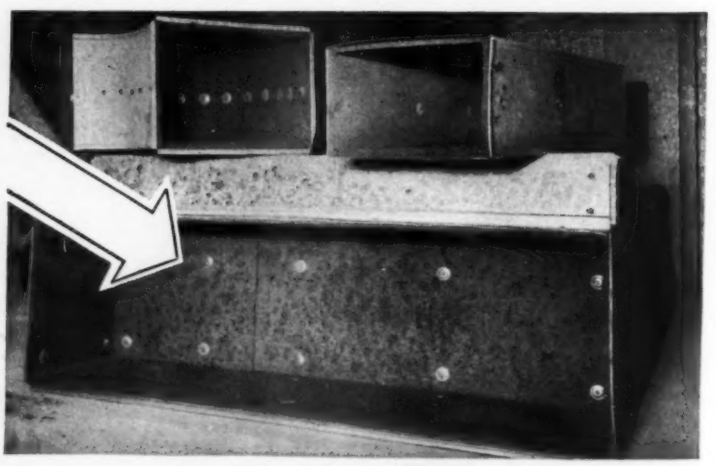
SERIOUS LOSSES have been reported in buildings where combustible sound absorbents—used to line air-conditioning ducts—have caught fire*.

The use of Johns-Manville Airacoustic Sheets for duct lining permanently eliminates this costly, unnecessary threat. Not only do these sheets provide as high as 80% sound absorption—but also, *they cannot burn or smolder. Can never contribute to fire spread.*

Furthermore, J-M Airacoustic Sheets are mineral in character . . . hence, permanent. Highly resistant to moisture . . . a serious enemy to the life of any material used in air-conditioning equipment. Light in weight, they are easily handled, quickly applied. Thermal conductivity is low; transverse strength high. And being smooth-surfaced—Airacoustic Sheets have a low frictional resistance.

For complete data on J-M Airacoustic Sheets, write Johns-Manville, 22 East 40th Street, New York City.

* In our files are bulletins from The National Board of Fire Underwriters, reporting several cases of this very nature.



JM Johns-Manville
AIR ACOUSTIC SHEETS FOR DUCT LINING

Baltimore branch of the Federal Reserve Bank of Richmond, Baltimore, Md.

MODERNIZE HEATING IN FEDERAL RESERVE BANK

Baltimore Bank, Constructed in 1927, Improves Service with Webster Moderator System

DISTRICT STEAM SERVICE

Baltimore, Md.—That heating systems sometimes become obsolete before they become old was demonstrated in the five-story building which houses the Baltimore branch of the Federal Reserve Bank of Richmond.

When constructed in 1927, the Federal Reserve Bank building was modern in every detail. In 1936, only eight years later, the rapid developments in heating science made the application of Webster Central Control a necessary step in keeping the heating system up-to-date.

With the Webster Moderator System, steam is delivered continuously to all radiators without excessive heating, thus correcting the distribution weakness of the old installation.

The need for a central heating control responsive to outdoor weather conditions has been met by an Outdoor Thermostat, which adjusts the steam supply automatically with every change in weather or wind direction. A manual Variator permits the operator to modify the effect of the thermostat.

"The Webster Moderator System has enabled us to maintain uniform, adequate temperatures throughout the building," says F. W. Wrightson, Assistant Cashier.

"There has been a noticeable improvement in the cleanliness of our building since the introduction of district steam service. This means better heating service."

The Baltimore Heat Corporation acted as modernization heating contractors.

Use of district steam for heating was authorized because of its economy and dependability in combination with the Webster Moderator System.

If you are interested in heating *new* buildings, or in improved heating service and lower heating cost in your present building, address

WARREN WEBSTER & CO., Camden, N. J.
Pioneers of the Vacuum System of Steam Heating
Representatives in 60 principal U.S. Cities—Est. 1888

WEBSTER MODERATOR SYSTEM OF STEAM HEATING "Controlled by the weather"
WARREN WEBSTER AND COMPANY CAMDEN, NEW JERSEY

The best of spokesmen for the Leland motor are those appliance builders whose products are Leland driven. They are in position to know, and knowing, speak.

In their considered opinion, The Leland is outstanding in design. It is well engineered and precision built. And, equally important, it is a motor on which they can get adequate and prompt field service wherever located.

Excellence of design—sound engineering—fine craftsmanship—dependable service—therein lies the explanation of Leland acceptance and Leland success.

THE LELAND ELECTRIC COMPANY

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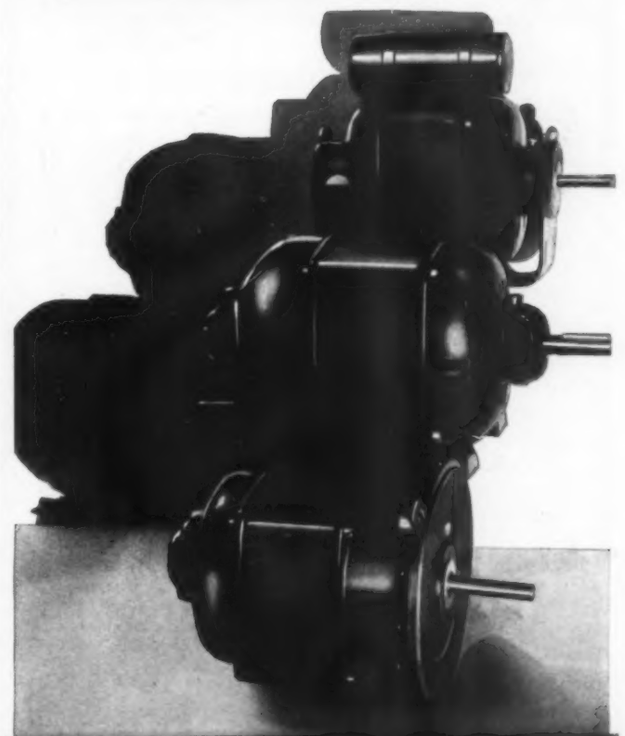

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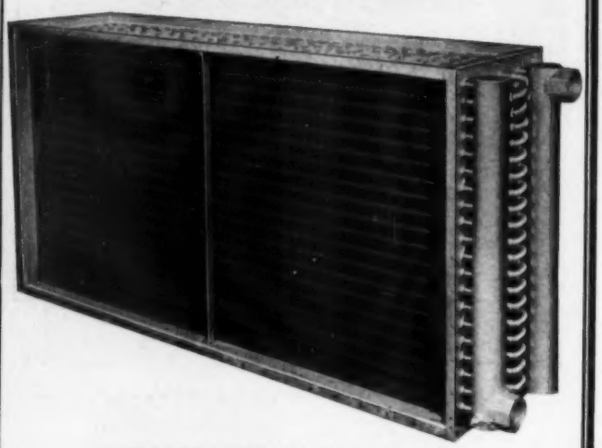
McQUAY

McQUAY HEATING AND COOLING COILS

For

All Applications Requiring Heating or Cooling of Air with Central Fan System.

HEATING—COOLING
AIR CONDITIONING



Air Conditioning Coils

Cooling coils for all Air Conditioning requirements using water, brine, or direct expansion refrigerants. Manufactured in an exceptionally wide range of established sizes and capacities to meet exacting requirements to assure a most economical and satisfactory installation. Continuous tube coils in all sizes up to 12 rows of tubes deep per section. Special coils built to specifications.

Steam Heating Coils

Durable high pressure construction for all applications. A large range of standard sizes to exactly meet all requirements and specifications. Dependable in every respect. Many large installations throughout the country. Approved by leading architects, engineers, and contractors. Descriptive catalogs furnished upon request.

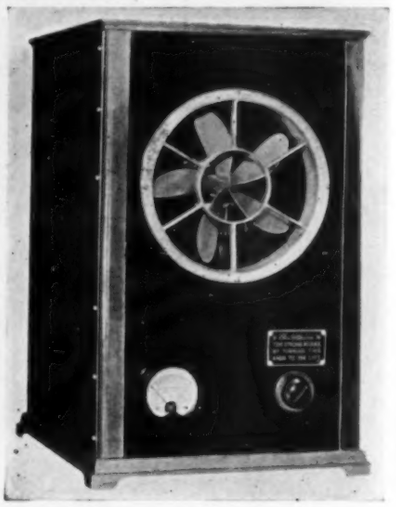
WRITE for New Catalogs on McQUAY Unit Coolers, Comfort Coolers, Cabinet Concealed Radiation, Refrigeration Coils, Ice Cube Makers, Air Conditioning Coils, Blast Coils, Combination Heating and Cooling Units. Suspended and Floor Type Blower Coolers, Cabinet Floor Type Room Coolers, Etc.

McQUAY inc
MINNEAPOLIS, MINN.

McQUAY

Ozone Generator Available in Two Sizes

No. 1224—New ozone generator, of modern appearance, is manufactured in two sizes, one of which is capable of ozonating 8,000 cu ft, and the other 20,000 cu ft. The panels are jet black bakelite trimmed in gleaming chromium-plated trim.



An improvement is the "Roto-Beam" circulator, which builds up a spiralling air stream which sucks in air from all sides and from ceiling to floor, states the maker. — Electroaire Corp., 1455 W. Congress St., Chicago, Ill.

Boiler Feedwater Control Announced

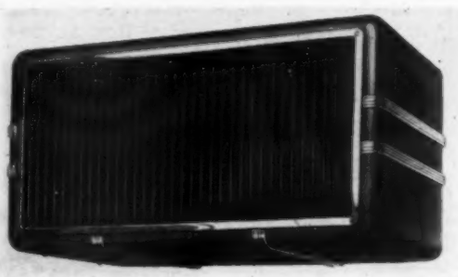
No. 1225**—Boiler feedwater level control to maintain levels within ½ in. is made in one size and will operate any valve up to 12 in. The illustration shows the controller mounted on a

fire-box boiler; a different mounting is provided for return tubular and water tube boilers.

It is called the "Campbell-Cantilever" level controller, and has many applications other than in boiler feed regulation, among which is the discharging of condensed steam from evaporators. —J. A. Campbell Co., Box 851, Long Beach, Calif.

New Comfort Cooler Line Announced

No. 1226—New comfort coolers, available in single fan and double fan models, have been designed particularly for installation as single units in small rooms or in multiple units for larger rooms where the construction of ducts is undesirable or impos-



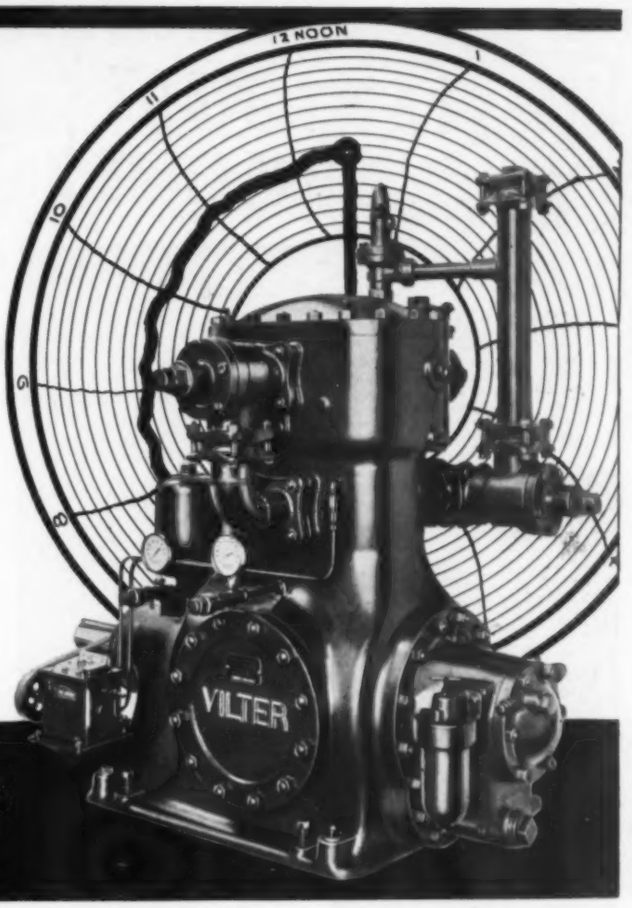
AMAZING HORSEPOWER *Economy* with a VILTER "FREON 12" COMPRESSOR!

Investment and subsequent cost of operation are vital factors in the prudent choice of any type of equipment.

By the elimination of many factors which require power, and by the refinement of others, the "VILTER FREON 12 COMPRESSOR" has shown amazing ability to deliver oversize performance at extremely low power cost.

A new and revolutionary shaft seal, plus other outstanding features contributing to efficient and economical performance with modern low-pressure refrigerants, make this "Freon 12" stand head and shoulders above the field in performance, dependability and economy.

It is another engineering achievement typical of VILTER. Let us tell you more about it.



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2148 South First Street
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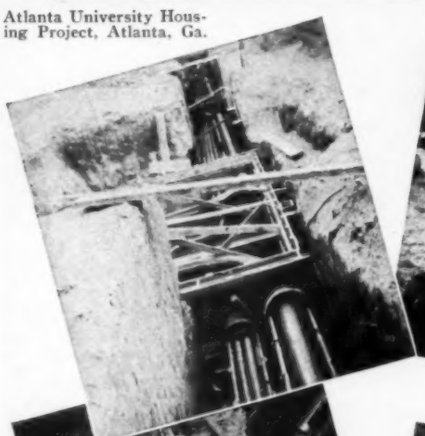
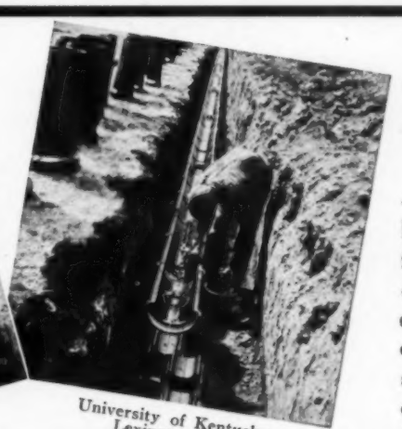
Atlanta University Housing Project, Atlanta, Ga.





University of Kentucky, Lexington, Ky.

Letterman General Hospital, San Francisco, Cal.

Moline Power Implement Co., Hopkins, Minn.

Thousands of Installations

—such as these—from coast to coast, bear witness to a national preference for Ric-wiL Underground Steam Conduit. Such confidence, on the part of engineers, architects, public officials, and owners, *must* be merited. Careful analysis of the basic features of Ric-wiL Tile or Cast Iron Systems, with incomparable Dry-paC Insulation, will convince *you* of Ric-wiL's better *investment value*. Write today for Installation Bulletin A3702.

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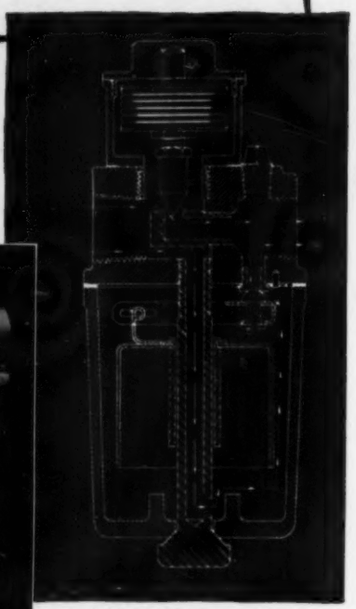
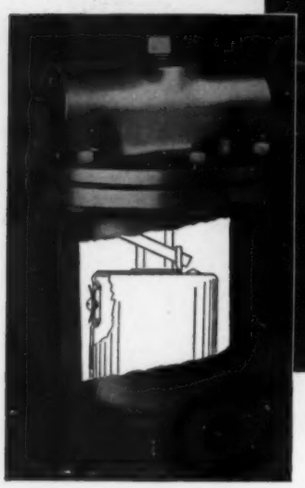
CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

INSTANT HEAT AT *any* PRESSURE



At the right is Anderson Super-Silvertop Combination Inverted Bucket and Thermostatic Trap. Notice how air is bypassed direct to return line. Gives instant heat on pressures from fraction of an ounce up to 50 pounds.



At theleft is Anderson Super-Silvertop Trap equipped with snap action Thermal Air Eliminator. For instant heat on pressures of two pounds or over.

● Instant heat at any pressure is another reason why Anderson Super-Silvertop is the only complete steam trap. Only with an Anderson Super-Silvertop can you get a combination inverted bucket and thermostatic bypass trap for use on pressures from fractions of an ounce up to 50 pounds. Only Anderson Super-Silvertop offers the snap action Thermal Air Eliminator for instant heat at pressures of two pounds and over.

Only with Anderson Super-Silvertop can you get *all* the advantages listed below.

Connect it either straight-in-line or as an elbow—no extra fittings to buy or install.

Is inspected, cleaned or repaired without disturbing any pipe connection.

Needs no strainer—handles ordinary dirt and scale.

Operates on vacuum as well as pressure, proof that it does not leak steam.

Averages 76% greater capacity than ordinary traps.

With snap action Thermal Air Eliminator it gives instant heat to the steam system.

Anderson Super-Silvertops are the greatest value today because they are complete and because they average 76% greater capacity. Send for complete information.

THE V. D. ANDERSON COMPANY
1942 West 96th Street • • Cleveland, Ohio

Anderson **SUPER-SILVERTOP**

sible. They are suspended type, propeller fan machines for cooling, dehumidification and air circulation, and may be had for direct expansion "Freon-12," methyl chloride, or circulating cold water, the refrigeration machine being remotely located.

The fan is a four blade, disc type, 16 in. diameter, rated at 600 cfm air delivery, driven by a 1/40 hp, 1000 rpm single phase, 60 cycle, 110 volt motor as standard equipment, other motors also being available. The cooling coil is constructed of copper tubes with metallically bonded copper fins. An air filter and filter frame can be supplied as optional equipment.—General Refrigeration Sales Co., Shirland Ave., Beloit, Wis.

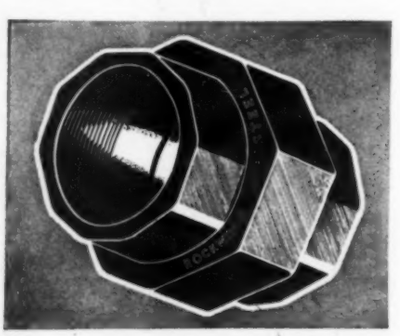
Evaporative Condenser Cuts Water Cost

No. 1227**—"Strang" evaporative condenser for air conditioning and commercial refrigeration service incorporates a patented feature, the submerging of the condenser to keep the entire condensing surface in contact with cool water, at close to the wet bulb temperature at all times, which is said to improve efficiency. The precooler, acting as a water eliminator while condensing, reduces the temperature of the refrigerant before it enters the submerged condenser proper due to heat exchange with the exhaust air after it has passed through the spray and to the evaporative cooling effect of free moisture entrained in the air stream which boils off the precooler surface, according to the maker.

Units are available for tonnage ratings from 1 to 45.—Motor Equipment Co., Wichita, Kansas.

High Strength Features Union

No. 1228—New "Dualsteel" union has been especially designed for service in oil refining and producing, rubber, chemical, bleaching and dyeing, shipbuilding, and paper plants, and, according to the maker, has increased strength and resistance to corrosion as well as the ability to withstand wrench abuse.



It is recommended by the manufacturer for 600 lb working steam pressure, 2000 lb hydrostatic pressure, and 900 F. It is

See how every grille bar can be set INDIVIDUALLY with

INDEPENDENT

"FABRIKATED" ADJUSTABLE DIRECTED AIR FLOW REGISTERS AND GRILLES

Air flows can be directed either up or down, right or left. The directional adjustment can be made at the time of installing, and after the system is operating, it may easily be changed at any time to make corrections necessary to meet unforeseen or changed conditions. Send for catalogs.

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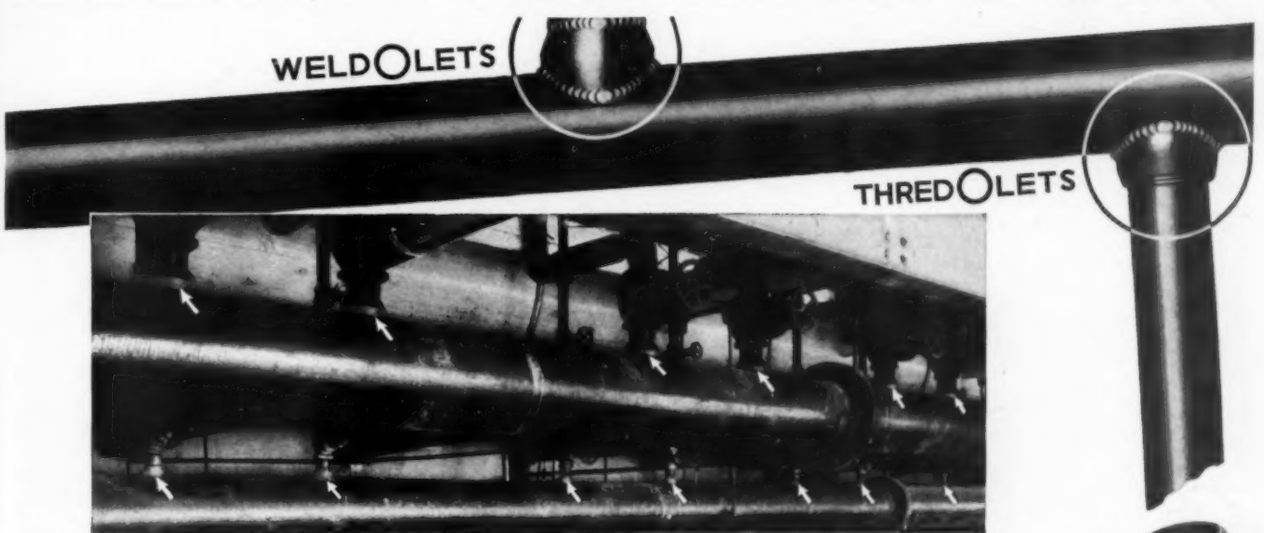


Get All the Advantages of Welded Piping

The installation of ThredOlets at the Union Bag & Paper Company's new plant at Savannah, Ga. (partially shown in the illustration) is typical of the jobs to which WeldOlets and ThredOlets are adapted.

No estimate of savings made on this job through their use compared with threaded fittings is available, but some idea of this saving is possible when it is realized that approximately 400 fittings were used without any cutting, forming, fitting or threading of the main pipe lines.

In addition, the ultimate in free flow conditions is achieved, full pipe strength is obtained at the junction, and leak-proof joints are obtained.

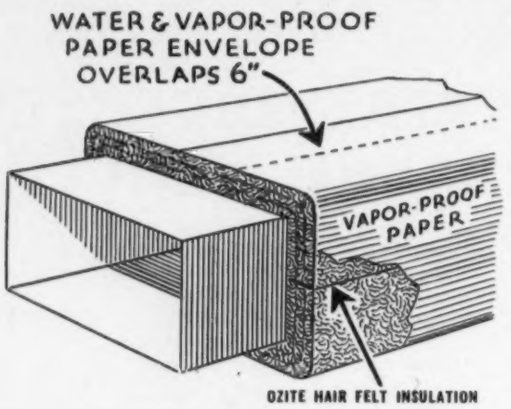
The cost of the fittings is no more, in many cases less, than ordinary fittings and by reason of the work eliminated, considerably less to install. ½" to 12" outlet sizes are available from stock to fit all standard pipe sizes. Bulletin WT21 tells the complete story. Write for your copy today.

"WeldOlets" and "ThredOlets" are fully covered by U. S. and foreign patents and are trade marks registered in the United States Patent Office.

BONNEY FORGE & TOOL WORKS Cedar & Meadow Sts., ALLENTOWN, PA.

WELDOLETS · THREDOLETS

Ozite HAIR FELT AIR DUCT INSULATION *KEEPS OUT HEAT!*



Leading engineers, architects and contractors specify Ozite Hair Felt Duct Insulation wherever utmost efficiency in an air cooling system is required . . . . *because:*

OZITE HAIR FELT DUCT INSULATION

has a thermal conductivity of only 0.246 B.t.u.'s . . . . reducing heat penetration to a minimum. It provides complete and lasting protection since it will not rot or decay. It is fire and moisture resistant and is repellant to vermin.

Ozite Hair Felt Duct Insulation is easy to apply, as it is flexible and readily shapes itself to the contours of the air duct . . . . fitting snugly around elbows and other sections that are usually difficult to insulate.

Write today for samples and complete specifications.

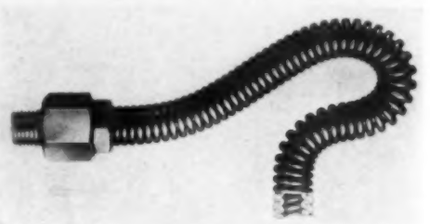
Ozite
HAIR FELT
AIR DUCT LINING
and
DUCT INSULATION
For All Sound, Heat, and Cold Insulation Purposes.

AMERICAN HAIR & FELT CO.
Merchandise Mart Chicago, Illinois

completely Parkerized, including the threaded area. May be had in sizes ½ to 2 in. inclusive.—Rockwood Sprinkler Co., 38 Harlow St., Worcester, Mass.

Flexible Metal Hose of New Construction

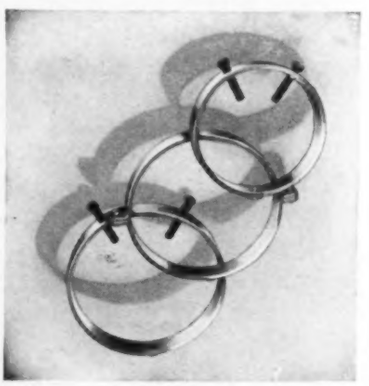
No. 1229—New flexible seamless metal hose (known as "T.S.T. Triple Safe Tubing"), instead of having exterior metal braid to take care of elongation under internal pressure, has the braid embedded between two corrugated seamless tubes to form an all metal, weldless, jointless and packless metal hose. This eliminates difficulties in cutting and attaching the braid, says the maker.



It is made of a special bronze alloy resistant to flexing and vibration, and can be made to order in aluminum. Standard sizes are ⅛, ⅜ and ½ in. I.D., and suitable fittings are available.—Seamlex Co., 5-19, 48th Ave., Long Island City, N. Y.

Announces Several New Products

No. 1230—Among the new products recently announced are pure wrought copper pipe hangers made especially for copper installations and in copper pipe sizes, which are supplied with threaded and notched supporting rod that can be cut to allow different lengths for proper pitch, facilitating installation. Ceiling plates are provided in round and oblong shape, the latter eliminating the cutting of a round plate when hanger must be attached next to beams or joists.



The same manufacturer has also developed a line of pre-fabricated air mains, complete with fittings and grilles. Another product is a new stream-line enclosure for convector type radiators, adaptable to both traditional and modernistic decoration, with horizontal discharge grilles.—American Radiator Co., 40 W. 40th St., New York, N. Y.

The coupon on page 160 is designed for your convenience in obtaining more information about any of these Equipment Developments.

Send for this NEW AUER Register Book!

THE latest AUER Register Book, just issued, is a complete presentation of the AUER Line for warm air heating and air conditioning. You will find this Book of genuine value in the selection of Registers, Cold Air Faces, and Grilles (including, of course, air directional registers.) The durability, efficiency, and beauty of AUER products need no comment. This illustrated Catalog is a handy reference book for *all* styles, with sizes and prices.

Write today for YOUR copy.

THE AUER REGISTER COMPANY, 3608 PAYNE AVENUE, CLEVELAND, OHIO

AUER DISTINCTIVE REGISTERS

& GRILLES For Air Conditioning *and* Gravity

ALL REFRIGERANT GASES ARE SAFE INSIDE THESE NON-POROUS FITTINGS

Because they're wrought copper like the pipe they join, trust Arco Full Flow Fittings to carry Freon . . . Methyl . . . *any* refrigerant. Requires no tinning . . . absolutely non-porous . . . safe against leaks . . . tight against vibration . . . Arco Copper-to-Copper makes a joint that's stronger than the pipe . . . that needs no tinning . . . that's easily made . . . that saves you money. Great variety of fittings from ¼" to 4".

Arco Pipe & Fittings Division
AMERICAN RADIATOR COMPANY
DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION
40A West 40th Street, New York, N. Y.

Patent Nos. 2,025,973, 2,005,969 and 2,002,470

ARCO

Full Flow WROUGHT
COPPER FITTINGS & PIPE
STOCKED BY JOBBERS EVERYWHERE

Booklets, Reports and Papers

Briefly Noted

Flow in a Pipe—The Use of an Elbow in a Pipe Line for Determining the Rate of Flow in a Pipe, by Wallace M. Lansford, Bulletin No. 289, Engineering Experiment Station, University of Illinois, Urbana, Ill. Price, 40c. Available without charge on application while supply for free distribution lasts.

Heat Transfer—Heat Transfer in Evaporation and Condensation, by Max Jakob, Reprint No. 10, Engineering Experiment Station, University of Illinois, Urbana, Ill. A series of lectures reprinted from *Mechanical Engineering*. Price, 35c.

The Construction Industry—Construction Volume II, the second of a series of three presenting statistics collected by the Census of Business in 1936 by a canvass of general contractors, operative builders, and special trade contractors. Includes data on employment by months, by occupational groups, by states and cities, and by kind of business. Issued by the Bureau of the Census, United States Department of Commerce. Copies available in limited quantities upon request.

Solar Energy—Solar Energy and Its Use for Heating Water in California by F. A. Brooks, Bulletin No. 602, Agricultural Experiment Station, University of California, Berkeley, Calif. Includes data on available solar energy of California and new experimental information on thermo-siphon circulation.

Warm Air Heating—Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems, dated February, 1937, issued by the *National Warm Air Heating and Air Conditioning Association*, 50 W. Broad St., Columbus, Ohio. Price, 50c.

Respiratory Dust Disease—Investigations on Respiratory Dust Disease in Operatives in the Cotton Industry, by C. Prausnitz, a report issued by the (British) Medical Research Council. Available from the British Library of Information, 270 Madison Ave., New York, N. Y. Price, 80c.

Fill Insulation—Use of Fill Insulation in the Construction of Refrigerated Rooms, by Willis M. Rees, published in the April, 1937, issue of *Agricultural Engineering*, journal of the *American Society of Agricultural Engineers*, St. Joseph, Mich.

Refrigerant Condensers—A Study of Heat Transfer in Unit Refrigerant Condensers Which Use Evaporative Cooling, by Byron E. James, read at the annual meeting of the *A.S.R.E.* and published in the March, 1937, issue of the journal of the *A.S.R.E., Refrigerating Engineering*.

"Trends" Summarizes Air Conditioning Research

The current issue of "Air Conditioning Trends" is devoted to a summary of some of the problems upon which research men are working, interprets them in terms which indicate their significance to the sales side of the industry.

The lead article features the views of A. E. Stacey, Jr., authority on process air conditioning, who calls the industrial air conditioning field a neglected market offering opportunities which have been lost sight of in the scramble for merchandise business. During the depression, the trend veered away from the industrial side, he says, and with the return of prosperity to many industries, this trend should reverse itself. The following are among the many industrial applications which could profitably be given more attention at this time, he believes:—

Precision machine tools—Controlled atmospheric conditions surrounding work involving clearances down to several thousandths of an inch.

Food products—Study of new products being manufactured. Bettering quality and flavor and adaptation to new manufacturing processes.

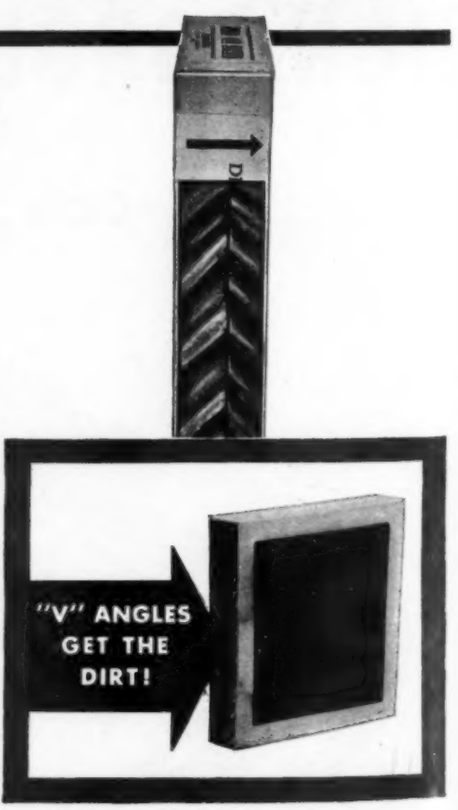
NO AIR CONDITIONING SYSTEM IS COMPLETE

WITHOUT THIS NEW

ARCO AIR FILTER

THE new Arco Air Filter for air conditioning and ventilating equipment makes other filters obsolete. Constructed with a special 90° "V" angle, it takes dirt—and *keeps* it. It holds more dust, lasts longer, keeps uniform low resistance to the end.

Arco Air Filters are light, odorless, and inexpensive. They're safe—won't drip oil at 180°, won't pass dry dust into the air stream. They're engineered on scientific principles and thoroughly tested. Make them standard on all your equipment. It pays! Write today for all the facts.



INDUSTRIAL DIVISION
AMERICAN RADIATOR COMPANY
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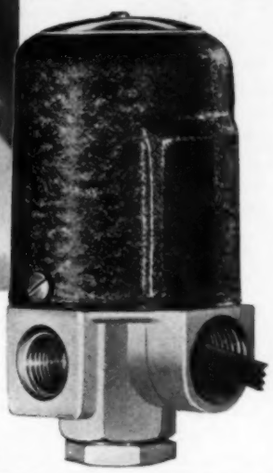
40 West 40th Street, New York, N. Y.

ARCO Air FILTERS

Easy to service



SOLENOID
CONTROL FOR AIR-CONDITIONING

Easy to disassemble and service. New accessibility has been added to an already proven efficiency in this NEW A-P SOLENOID. Easily and quickly disassembled for inspection and cleaning.

Insist on A-P Solenoids for your air-conditioning equipment. Write for Bulletin on #73-RB.

AUTOMATIC PRODUCTS COMPANY
2460 NORTH THIRTY — SECOND STREET
MILWAUKEE WISCONSIN

These words ended a sentence in a recent letter to the Pomona Pump Co. The paragraph read. "I've pumped a lot of water in my time and used about every make of pump to do it, too. For the last 10 years I've been buying Pomonas. Sometime back I read an ad that said, 'Look At All Three'. Thought I'd do it, getting another pump, and so I bought another Pomona."

Five short words, but they just about sum up the universal attitude of Pomona owners. The most highly organized engineering department in the turbine pump industry, aided by complete research and testing departments is the reason why all the world looks to Pomona.

POMONA PUMP CO.
Manufacturing Plants: Pomona, California - St. Louis, Missouri
Sales Offices: San Francisco, New York, Chicago, Los Angeles

POMONA TURBINE PUMPS

Welding fittings at their best

WELD your pipe lines—for added safety and added strength with lighter weight — for smooth-flow interiors — for smooth, sightly, easily insulated exteriors. Taylor WeldELLS head the list of Taylor Forge *Engineered* Welding Fittings. They are reinforced at the crotch where strain is greatest—have tangents which remove the weld from the curved section—have machine tool bevelled welding surfaces which permit sound, dense welds.

The Taylor Forge line covers every need. It is the most complete line made.

TAYLOR FORGE & PIPE WORKS
General Offices and Works:
P. O. Box 485, Chicago,
New York Office: 50 Church St.
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Also made from BYERS GENUINE WROUGHT IRON PIPE

TAYLOR FORGE
WeldELLS
SEAMLESS PIPE FITTINGS FOR WELDING

AIR CONDITIONING NOZZLES

Nozzles of unique Yarway Involute Design.

No internal parts or vanes to clog or erode.

Insure trouble-free air washing service.

Sizes and types for all requirements.

Many large users—installations total more than 5 million gallons per minute in water cooling and air conditioning service. Write for Bulletin N-615.

YARNALL-WARING COMPANY
MERMAID AND ANDERSON STS.
PHILADELPHIA

YARWAY

No. 17 Beaver 1/8 to 2-Inch

Sold in units! Buy any range! The large holes for oiling and chip clearance are bridged across the top forming an arch between the dies—which prevents spreading of die slots. Die segments are square in shape—no weak "offset" section to develop water cracks in heat treating. Special long-throat die grinding for easy starting and easy pulling.

BEAVER PIPE TOOLS

537 KEENEY AVE. WARREN, OHIO

OILS are getting heavier. The No. 6 oil of today is a far more viscous fluid than the No. 6 of yesterday. To meet the more severe control requirements of these heavy oils, the new General Controls K-10 Quiet Solenoid has been designed with added power in the solenoid and a more powerful leverage action.

Put on a line carrying a No. 6 oil at low temperature, with a circuit-breaker bringing the solenoid on and off 200 times an hour, this new K-10 was subjected to a grueling test of one million continuous operations without failure. If the oil will flow, this valve will control it satisfactorily.

$11.00 LIST PRICE

GENERAL CONTROLS

1503 Broadway, Cleveland, Ohio
267 5th Ave., New York City, N.Y.
1370 Harrison St., San Francisco, Calif.
421 Dwight Building, Kansas City, Mo.

Knitting mills—Production conditions have changed materially over past practices. Effect of air conditioning on rate of production and individual efficiency of employees.

Foundries and asbestos working plants, grinding rooms and certain mining operations, where better knowledge of how to eliminate or minimize hazardous dusts is needed and where united action, through more intensive cooperative research, will be profitable.

Walter L. Fleisher's views on the importance of research on the health aspects of air conditioning are given in the report of an interview. Air conditioning engineers are approaching a working union with the medical profession that is likely to lead, Mr. Fleisher believes, to a vast new field for practical applications of air conditioning in the field of therapy.

Other stories are devoted to the influence on future developments in air conditioning and the opening of larger markets by research sponsored by the *A. S. H. V. E.*, the importance of research in railway air conditioning, and studies on the insulation of roofs and methods of reducing cooling load. Dr. C. A. Mills makes the interesting prediction that it will be profitable to provide some form of air conditioning for livestock, to improve the quality of fine-bred race-horses, make cows give more milk, hens lay more eggs.

"Trends" is published by Keeney Publishing Co., 6 N. Michigan Ave., Chicago, Ill. Copies are available upon request on business letterhead.

Recent Trade Literature

For your convenience in obtaining copies of bulletins, see coupon on page 160.

No. 2339. AIR CONDITIONERS: Pleasantaire Corporation, 304 E. 45th St., New York, N. Y. 8 p. bulletin on "Northwind" ¼ ton portable summer air conditioners for window installation, with "Freon" or methyl chloride ⅓ hp compressor and 650 cfm fan.

No. 2340. AIR CONDITIONERS: Utica Radiator Corporation, 2201 Dwyer Ave., Utica, N. Y. 16 p. bulletin on washed air conditioning for offices, stores, factories and homes, describing standarized unit and its features of construction and discussing its applications and installation; also a trade price sheet giving specifications and prices.

No. 2341. AIR CONDITIONING: Westinghouse Electric & Mfg. Co., 200 E. 5th St., Mansfield, Ohio. Data sheets for air conditioning application manual covering condensing units, compressor and condenser performance data, evaporators, self-contained air conditioning units, and heat transfer units.

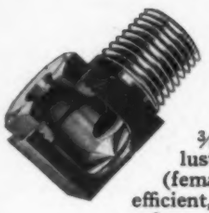
No. 2342. AIR CONDITIONING PRODUCTS: Peerless of America, Inc., 515 W. 35th St., Chicago, Ill. Loose-leaf catalog of air conditioning products including floor and ceiling type units, expansion valves, heat exchangers, evaporative condensers, and various refrigeration products. Specifications and capacity data are given.

No. 2343. AIR CONDITIONING UNITS: Clarage Fan Co., Porter St., Kalamazoo, Mich. 4 p. bulletin on "Multitherm" air conditioning units for cooling, dehumidifying, heating, humidifying and cleaning, available in 988 equipment combinations. Dimensions and physical data are tabulated and features are described in detail.

No. 2344. BOILER FEED PUMPS: Worthington Pump & Machinery Corp., Harrison, N. J. 4 p. bulletin on balanced multistage volute centrifugal pumps for boiler feed service up to 1200 lb per sq in., capacities from 200 to 1500 gpm.

No. 2345. BOILERS: American Radiator Co., 40 W. 40th St., New York, N. Y. 8 p. bulletin on "Ideal" boilers for automatic coal firing, giving general information on their selection

MONARCH
non-clogging
SPRAY NOZZLES



For Air Washing

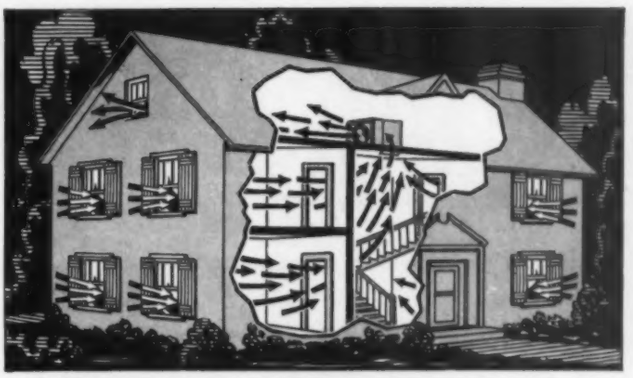
We recommend either ⅜" Fig. 629 male pipe (illustrated) or ¼" Fig. 631 (female) Brass nozzles. Small, efficient, and inexpensive, they produce an evenly distributed hollow cone spray. One large lead hole minimizes any clogging tendency. Capacities from 4.7 G.P.H. up at 40 lbs.

For Re-Cooling

Monarch Fig. B-8 and new B-8-A cast Brass nozzles, for cooling condensing water in spray ponds, use no internal vanes or deflectors which might facilitate clogging. Available in capacities up to 104 G.P.M. at 10 lbs.

Write for New Catalog 6-A

MONARCH MFG. WORKS, INC.
2728 E. Westmoreland Street, Philadelphia, Pa.



Hidden Treasure
IN UNVENTILATED ATTICS

Thousands of Homes are in the market for National Home Ventilators. There are big profits awaiting the man who anticipates that demand and is prepared to sell them.

NATIONAL

"Air Flow" Home Ventilator

Installed in the attic, it exhausts heated daytime air from entire house, drawing in cooler night air through doors and windows. Built in 24", 30", 36", 42", and 48" sizes. This will be a fast seller.

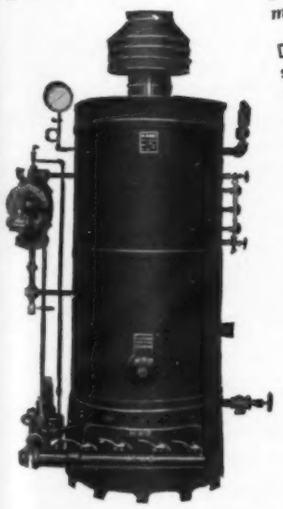
100% AGENCY FRANCHISE

Write today for the National catalog showing complete line of blowers, washers, fans, coils, etc., and full information on our exclusive franchise. State territory desired.

NATIONAL FAN AND BLOWER CORP.
540 W. WASHINGTON BLVD. CHICAGO, ILL.

LOOKING FOR A WAY TO CUT COSTS?

Then install the KANE Automatic Gas Fired Steam Boiler!



Dependable in operation . . . supplying always clean steam under constant pressure . . . the KANE is entirely automatic in action. Thus, it requires the very minimum manual attention (reduces labor costs), and, because it burns gas only when steam is needed, the KANE holds fuel costs to an irreducible minimum.

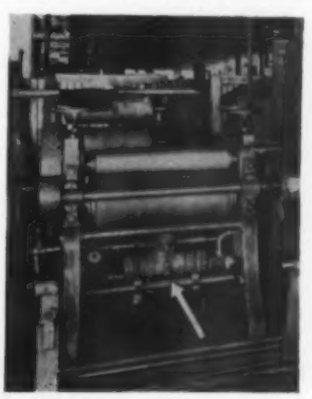
Judged by records of actual performance under varied, oftentimes difficult conditions in the field, the KANE soon pays for itself in labor, maintenance and fuel savings . . . the chances are it can cut your costs, too.

May we tell you more about the KANE? Details will be mailed you promptly upon request.

MEARS-KANE-OFELDT INC.

Executive Office and Factory
1903-1915 EAST HAGERT ST., PHILADELPHIA
Branch Offices or Distributors in Principal Cities
Also makers of the well-known Ofeldt Water Tube Gas Steam Boiler and M-K-O Automatic Boiler Feed Pump

Inexpensive
electric
steam boiler permits of shutting down the main steam plant—



CHROMALOX
ELECTRIC HEATING UNITS
save money in summer operation

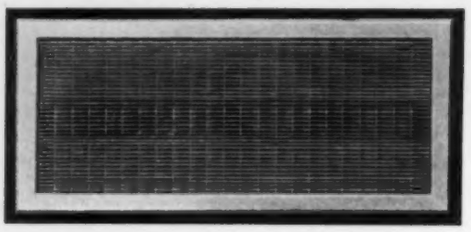
An arrangement of pipe fittings in which is inserted a Chromalox immersion unit, provides all the steam needed to supply the heated rolls in the paper machine illustrated. This makes unnecessary the operation of the main steam plant, geared up to winter heating needs and providing summer process steam at an excessive cost.

The small, electrically heated "boiler" is furnished with automatic pressure control, which turns the electric heat on or off as required. No make-up water is needed nor attention for other reasons.

There is a vast field in industry for the application of handy, economical electric heat to meet conditions such as those described. Heating Contractors are invited to avail themselves of the aid and experience of Wiegand engineers to solve any specific problem. Your request brings the Chromalox Book of Electric Heat.

EDWIN L. WIEGAND COMPANY
7610 Thomas Blvd. Pittsburgh, Pa.

WATERLOO AirMaster GRILLES



Engineered scientifically for accurate, controlled diffusion and direction of incoming air. The exterior bars are set at specified angles; interior supporting bars adjustable individually left and right (number of settings unlimited); 74.4% free area; no raw edges exposed to air flow. New straight-line design harmonizes perfectly with modern decorating treatments. Full details mailed on request. Write.

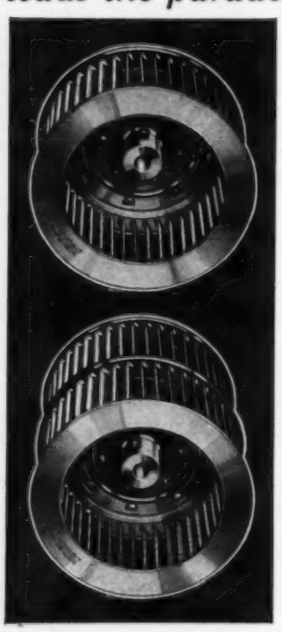
Waterloo Register Co.
Waterloo, Iowa—Seattle, Wash.
In New York City—Air Conditioning Utilities
8 W. 40th St.

TORRINGTON EQUIPMENT *leads the parade*

Backed by broad experience in the industry, TORRINGTON engineering is years ahead — is leading the way to increased dependability and better quality. Consider these points and what they can mean to your product.

- Smoother, quieter operation.
- Non-rusting, non-resonant construction
- Lower starting torque
- Lighter in weight yet rugged
- Less power consumption
- Every wheel hand inspected and statically balanced

Write for information about our improved line of propeller fans. Also ask for our catalog of blower wheels with ratings for all sizes and table of dimensions for housing scroll design.



TORRINGTON MFG. CO. TORRINGTON CONN. U.S.A.
TORRINGTON FANS AND BLOWER WHEELS

and application, and a 6 p. bulletin giving rating data for handy reference and quick selection.

No. 2346. BOILERS AND RADIATORS: U. S. Radiator Corp., 1056 National Bank Bldg., Detroit, Mich. Price change notice on "Capitol" boilers, radiators, convectors and enclosures.

No. 2347. BRAZING ALLOYS: Handy & Harman, 82 Fulton St., New York, N. Y. 4 p. bulletin describing "Easy-flo" brazing alloy for ferrous and non-ferrous metals which is liquid and fast flowing at 1175 F.

No. 2348. CABLE: General Electric Co., 1 River Rd., Schenectady, N. Y. 4 p. bulletin of tips on the uses of electrical cable of various types, illustrating and describing features and applications.

No. 2349. COMPRESSORS: Carrier Corp., 850 Frelinghuysen Ave., Newark, N. J. 4 p. bulletin on centrifugal compressors for new plants, additions, and replacements, listing 15 applications and discussing five factors that produce economy and efficiency. A partial list of users is given.

No. 2350. COMPRESSORS: Worthington Pump & Machinery Corp., Harrison, N. J. 18 p. bulletin on opposed steam driven air compressors for heavy duty.

No. 2351. CONDENSING UNITS: Uniflow Mfg. Co., E. Lake Rd., Erie, Pa. 4 p. bulletin on air and water cooled condensing units, 1/6 to 10 hp, with specification and capacity tables.

No. 2352. CONTROLS: Automatic Products Co., 2450 N. 32nd St., Milwaukee, Wis. Loose-leaf catalog of controls for heating, air conditioning, ventilating and refrigeration, including humidistats, thermostats, solenoid refrigerant valves, solenoid water valves, thermostatic expansion valves, metering valves, oil control valves, relays, and other items.

No. 2353. CONTROLS: Barber-Colman Co., 225 Loomis St., Rockford, Ill. 96 p. catalog describing and explaining controls and control applications for various types of air conditioning systems. Subjects discussed include ventilation only—summer and winter operation, methods of controlling fresh air and recirculated air, year round air conditioning systems of the surface cooler and air washer types, control of refrigeration, unit and cabinet coolers and air conditioners, and variable control of indoor temperature. Technical information on this company's air conditioning controls is given and many diagrams of control hook-ups are shown.

No. 2354. CONTROLS: Mercoid Corp., 4201 Belmont Ave., Chicago, Ill. 16 p. booklet on automatic controls, including thermostats, safety controls, limit and low water controls for steam boilers, hot water temperature limit controls, warm air furnace controls, and mercury contact switches, briefly describing the features and applications of the units, and with illustrations of each.

No. 2355. DUST ARRESTERS: Northern Blower Co., 6409 Barberton Ave., Cleveland, Ohio. 4 p. bulletin describing the "Norblo" bag type dust arrester and giving dimensions and capacity tables.

No. 2356. EVAPORATIVE CONDENSERS: Motor Equipment Co., Wichita, Kans. 4 p. bulletin describing the "Strang" evaporative condenser, and a 12 p. bulletin of technical data on these units, discussing constructional details and giving rating tables and installation diagrams.

No. 2357. FURNACE WALL ARCH CONSTRUCTION: Geo. P. Reintjes Co., 2517 Jefferson, Kansas City, Mo. 4 p. bulletin on how simplified arch construction improves furnace structures, illustrating and describing the features of this company's method.

No. 2358. GRILLES AND REGISTERS: Tuttle & Bailey, Inc., Corbin Ave., New Britain, Conn. 56 p. catalog of grilles, registers and air control devices for air conditioning, with a synopsis and index which permits easy reference. Each of the many types is described and numerous air capacity tables and other technical data are given. Many types of units are included, among them being various grille designs, high velocity

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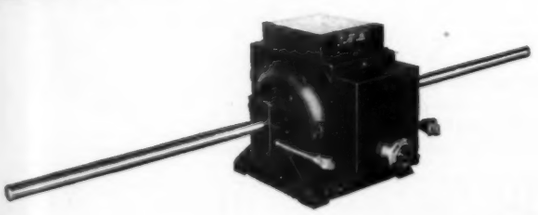

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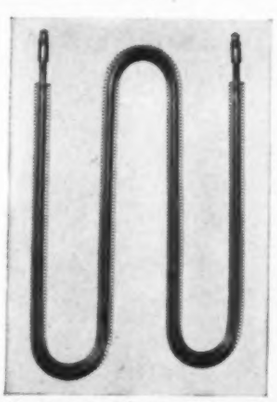

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No. 2360. INDUSTRIAL MOUNTINGS: U. S. Rubber Co., 1790 Broadway, New York, N. Y. 24 p. booklet on "U. S. Royal" industrial mountings for absorbing vibration, describing and diagramming various types of installations, and including pertinent data.

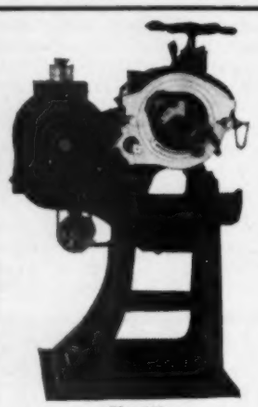
No. 2361. INSULATION: Insulite Co., 1100 Builders Exchange Bldg., Minneapolis, Minn. 24 p. booklet on roof insulation, giving specifications for various types, data on prevention of condensation, thermal coefficients for various types of roof construction, a fuel savings analysis of a typical industrial roof, etc.

No. 2362. OIL BURNERS: Simplex Oil Heating Corp., 30 Church St., New York, N. Y. 4 p. bulletin describing horizontal rotary type oil burners for capacities of 4, 10, 25, and 40 gallons per hour in fully automatic, semi-automatic or manually operated types.

No. 2363. PROCESSING EQUIPMENT: Edge Moor Iron Works, Inc., 30 Rockefeller Plaza, New York, N. Y. 8 p. bulletin illustrating fabricated processing equipment for various industries, and describing the facilities of this manufacturer of such equipment.

No. 2364. PROCESSING EQUIPMENT: International Nickel Co., Inc., 67 Wall St., New York, N. Y. Issue of "Process Industries Quarterly" (Vol. 1, No. 4) including items on various processing equipment in different industries.

No. 2365. PUMPS: American Steam Pump Co., 60 Capitol Ave., N. E., Battle Creek, Mich. 4 p. bulletin on "American-Marsh" turbine pump with only one moving part, describing



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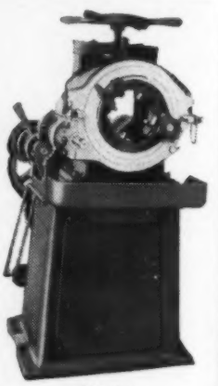

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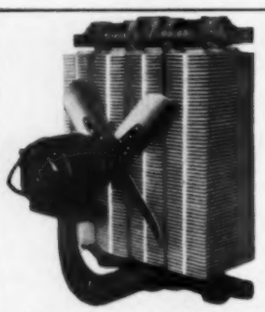

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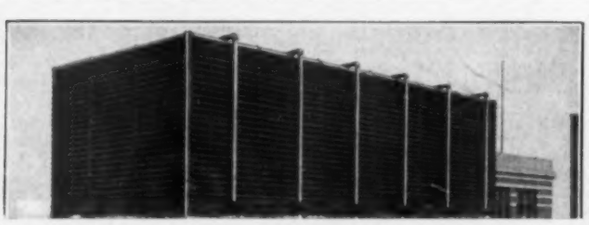

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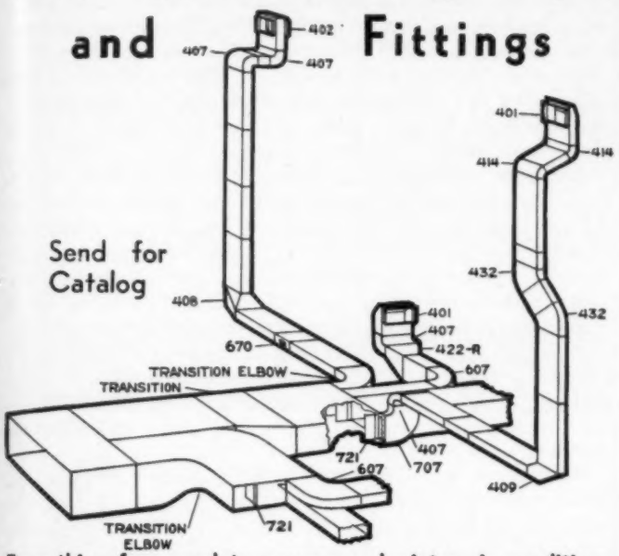

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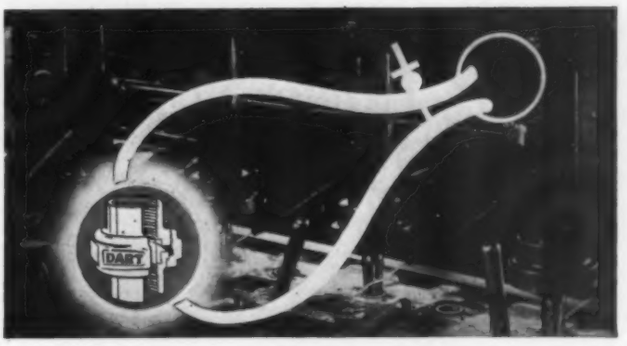

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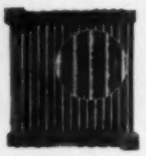
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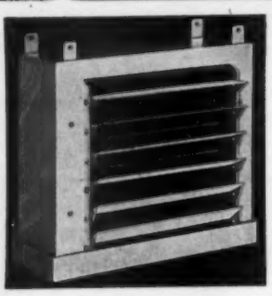

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No. 2366. PYROMETERS: Brown Instrument Co., Div. of Minneapolis-Honeywell Regulator Co., 4534 Wayne Ave., Philadelphia, Pa. 40 p. catalog of millivoltmeter type pyrometers—indicating, recording, controlling, multiple recording, and multiple controlling—illustrating and describing the various types and their features in detail, and suggesting various applications.

No. 2367. RADIATOR ENCLOSURES: American Radiator Co., 40 W. 40th St., New York, N. Y. 4 p. bulletin illustrating the adaptability of "Arco" radiator enclosures to the interior decoration of rooms.

No. 2368. RECORDERS AND CONTROLLERS: Bristol Co., Platts Bridge, Waterbury, Conn. 8 p. bulletin on new line of low range pressure and draft recorders and controllers.

No. 2369. RECORDING INSTRUMENTS: General Electric Co., 1 River Rd., Schenectady, N. Y. 32 p. booklet on strip chart electrical recording instruments for industrial and public utility uses. Many types are included.

No. 2370. RECORDING INSTRUMENTS: Practical Instrument Co., 2717 N. Ashland Ave., Chicago, Ill. 4 p. folder describing portable recording thermometers and electrical operation recorders for use in refrigeration, air conditioning, and heating.

No. 2371. THERMOMETERS: American Schaeffer & Budenberg Div., Consolidated Ashcroft Hancock Co., Inc., 11 Elias St., Bridgeport, Conn. 46 p. catalog of glass thermometers which have a number of features, including red reflecting glass, monel metal bulb chamber, highly polished chrome plated frame, and individually calibrated scales. Features are described in detail and many types illustrated.

No. 2372. UNIT VENTILATORS: Peerless Unit Ventilation Co., Inc., 810 Union Ave., Bridgeport, Conn. 12 p. bulletin on unit ventilators and heating and ventilating units, describing construction and control and giving diagrams of various applications and dimensions, capacity tables, and specifications.

No. 2373. VACUUM HEATING PUMPS: Nash Engineering Co., South Norwalk, Conn. 8 p. bulletin on "Jennings" manifold type vacuum heating pump for return line vacuum heating systems, or any service where a liquid and a gas are handled together. Features are described and capacity tables and dimensions given.

No. 2374. VALVE DATA: General Electric Co., 1 River Rd., Schenectady, N. Y. Loose-leaf valve data book giving complete information on construction, operation and application of electrically operated valves, including rating tables, diagrams, photographs, and information simplifying selection and application.

No. 2375. VIBRATION: Felters Co., Inc., 210 South St., Boston, Mass. 32 p. booklet on the study of vibration in plant

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No. 2376. VIBRATION ELIMINATORS: American Metal Hose Branch, American Brass Co., 414 Meadow St., Waterbury, Conn. 4 p. bulletin describing flexible vibration eliminators for refrigerant lines for air conditioning and refrigeration installations consisting of seamless flexible bronze tubing with standard copper tube slip fittings welded to both ends.

No. 2377. VIBRATION ELIMINATORS: Vibration Eliminator Co., 41-26 37th St., Long Island City, N. Y. 20 p. booklet on stopping vibration and isolation of machinery, describing improved vibration eliminators, the use of natural cork plates, and ceiling suspension vibration eliminators. Various installations under actual working conditions are illustrated.

No. 2378. WATER RADIATORS: Star Radiator Co., 649 Ceres Ave., Los Angeles, Calif. 12 p. bulletin of performance data for "Sta-cool" water radiators for air conditioning, industrial engine cooling, and heating and cooling of air and water; also a 12 p. bulletin on supplies and equipment for use with these radiators.

No. 2379. WELDERS: Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. 8 p. bulletin on "Flexarc" motor driven welders and bare generators, with information on application, construction, specifications and performance, for single operator 200, 300, and 400 ampere units; also a similar bulletin on engine driven welders.

No. 2380. WELDING: Lincoln Electric Co., 12818 Coit Rd., Cleveland, Ohio. 12 p. bulletin entitled "Building a Career in Arc Welding" describing the practical and technical courses of this company with illustrations of facilities available for training in welding.

No. 2381. ZONED HEATING SYSTEMS: Webster Tallmadge & Co., Inc., East Orange, N. J. 8 p. bulletin on zoned heating systems describing their advantages and illustrating and listing installations. Also a 4 p. bulletin on motorized on and off valves and heating system devices.

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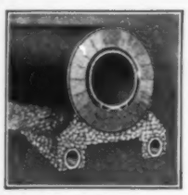

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